STEAM ENGINE INDICATORS AND VALVE GEARS

A PRACTICAL PRESENTATION OF MODERN TESTING APPLIANCES AND METHODS USED TO PRODUCE MAXIMUM EFFICIENCY AS APPLIED TO THE STEAM ENGINE

BY

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INTRODUCTION

JAMES WATT was responsible for many important developments in connection with the steam engine and one of these was the "Indicator Diagram". By means of this ingenious graph of the engine's action a trained engineer can determine its ailments as surely as a skilled physician can detect the weaknesses of a patient's heart action by the aid of a stethoscope. Every deviation of the curve from the standard form means to this expert a fault either of design or of adjustment. Poor lubrication, late admission of the steam, excessive back pressure, too early cut-off, etc., each makes its impression on the curve, and each trouble in turn can be corrected and proof given that this has been done by noting the improvement in the curve on a new indicator card.

¶ In addition to this information, a measurement of the area of the diagram, together with known constants of the engine and indicator, enable one to determine the exact number of horsepower produced by the engine.

¶ Another important adjunct of the modern engine is the "Valve Gear", by which the admission of the steam to the cylinder, the cut-off, the expansion, compression, and exhaust are controlled. The proper operation of the valves of an engine is of the highest economic importance and not only must the expert engineer understand the working theory of this control device and understand the differences between a Stephenson, Walschaert, or Reynolds-Corliss, for example, but he must be able to determine whether the valve actions are as perfect as they can be made by proper adjustment. By use of a graphical method called a "Zeuner Diagram", it is possible to determine the proper lap, lead, angle of advance, cut-off, and release, and to correct any errors of adjustment that may exist.

¶ All of these important matters in connection with the steam engine are carefully and authoritatively treated in this book in an exceedingly practical way. A number of examples taken from actual operation experiences are carefully worked out as a guide to the proper method of applying both the indicator and Zeuner diagrams.

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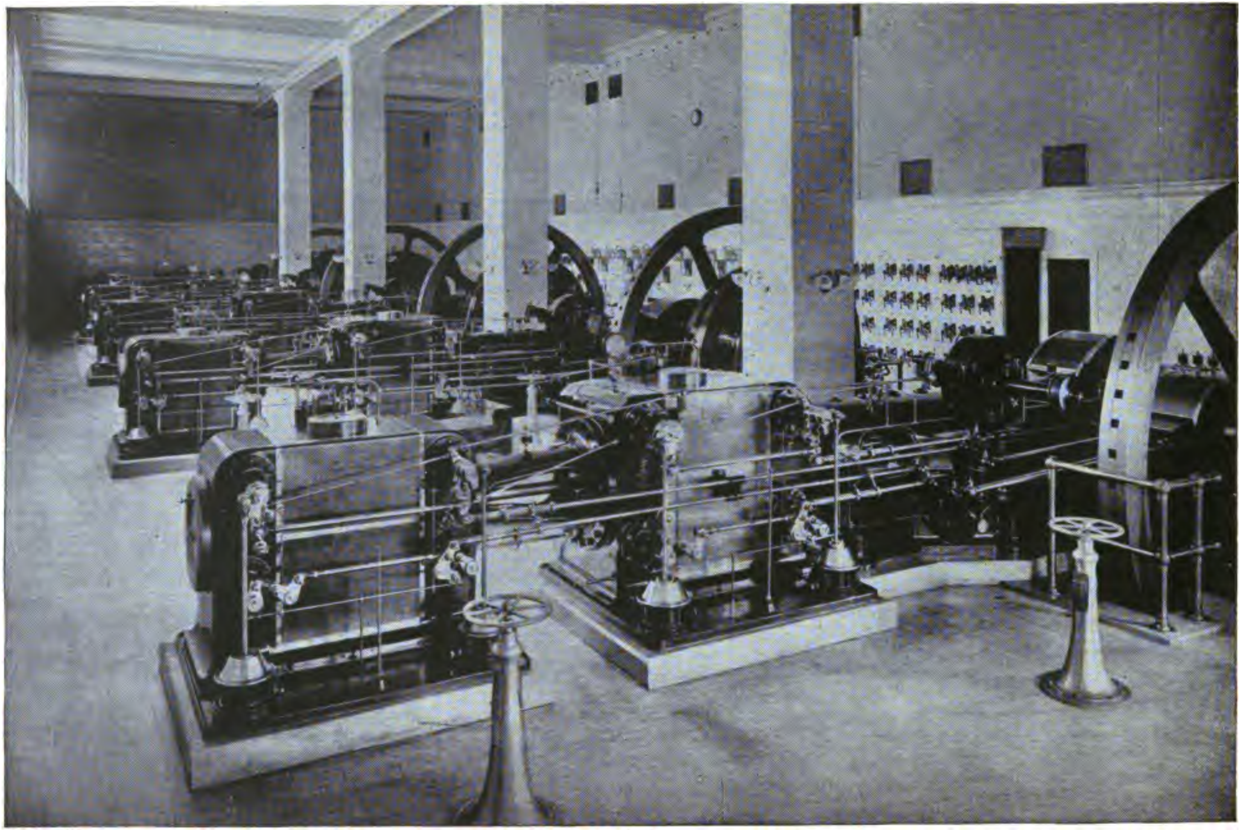


FIVE TANDEM-COMPOUND NON-CONDENSING RICE AND SARGENT CORLISS ENGINES OPERATING DIRECT-CURRENT GENERATORS IN PARALLEL

Courtesy of Providence Engineering Works, Providence, Rhode Island

PART I

STEAM ENGINE INDICATORS

INTRODUCTION

The steam engine indicator is an instrument designed to make an accurate graphical diagram of the pressure of the steam in the engine cylinder at all points of the stroke. This diagram affords a means for studying the performance of the steam engine.

The indicator serves two very important purposes, although many other results are obtained by its use. (1) *In the hands of an experienced engineer, it enables him to discover any defects in the design or setting of the valve mechanism.* (2) *It also indicates whether the steam ports are large enough and, in fact, a proper interpretation will disclose the exact condition of the design and operation.* Thus the engineer can determine whether any change in the operation of the moving parts is advisable.

The information that may be obtained by an intelligent use of the indicator is of very great value to the engineer. The power of the engine at any time and under any condition may be determined; many facts can be accurately obtained that could not be secured in any other way; many things about the steam engine that before seemed mysterious are now made clear. Its value is so universally recognized that almost all builders of steam engines apply indicators to their engines and adjust the valves and moving parts before sending the engine away from their factories. For these and other reasons which might be mentioned, it is seen that the indicator has played no small part in the development of the steam engine.

In the early development of the steam engine by James Watt, he realized that some means should be provided whereby the internal action of the steam, valves, etc., could be watched or their behavior interpreted. As a result of this apparent need, the indicator was devised, the first forms being crude in their construction but the underlying principles being the same as are found today in the modern instrument. It is, therefore, of interest to note that the changes

made in the indicator since its advent have been largely in constructional details rather than in principle. The moving parts of the earlier indicators were exceedingly heavy; on this account, the inertia of the moving parts often distorted the indicator diagram to such a degree that the results obtained were unreliable. The older types would give fairly accurate results on slow-speed engines but were useless on high-speed engines on account of the comparatively great weight of the pencil mechanism and other moving parts.

The modern indicator is almost perfect in construction. All of its parts are as light as good design will permit and it is conveniently manipulated and easily adjusted. It may and does at times, however, record pressures incorrectly. Some of the most common errors, which are often misleading, will be discussed later.

In order to have an intelligent understanding of the use and care of an indicator, it is necessary to become familiar with its construction, and to that end, a description of three well-known makes will be given, viz, the Crosby, Tabor, and Thompson.

TYPES

It will be observed that indicators do not differ in detail very materially, their chief difference being found in the pencil mechanism. In order to make a discussion of the construction logical in development, it is well to note first an improved form of the Watt indicator.

Watt Indicator. The Watt indicator, Fig. 1, consists of a steam cylinder *S*, about 1 inch in diameter and 6 inches long, in which a solid piston *P* is accurately fitted. A spiral spring *A* is attached to this piston, and controls the motion of a pencil *a*, which is also attached to the piston. This pencil can operate on a sheet of paper fastened to a sliding board *B*. This board moves back and forth by means of a weight at one end and a cord at the other which is connected to some reciprocating part of the engine. The indicator cylinder *S* may be put in communication with the engine cylinder by means of the cock *C*. With this instrument, a complete diagram can be taken. When cylinder *S* is put in communication with the engine cylinder by means of the cock *C*, pencil *a* is raised or lowered precisely as the intensity of the pressure in the cylinder varies. This variation of height of pencil or pressures is registered upon a card

carried by the board *B*. As the board *B* is moved in exact coincidence with the piston of the engine, by being connected to some reciprocating part, the resulting card gives an exact indication of the pressure in the cylinder for all points of the stroke. The vertical dimensions of the card, commonly called the *ordinates*, indicate the pressures; the horizontal dimensions, or *abscissas*, indicate the simultaneous positions of the piston.

It is a natural transition from the earlier form shown in Fig. 1 to the modern indicator, as the intervening changes have been largely in the perfection of the recording mechanism and in the refinement of details, as will be pointed out later.

Crosby Indicator. The Crosby indicator is illustrated in cross-section in Fig. 2. The indicator cylinder *4* is connected to the steam engine cylinder by means of the loose nut *7*. The steam passes from the steam engine cylinder to the indicator cylinder through the passage *6* and acts on piston *8*. The indicator cylinder is very carefully designed and constructed so that the piston will have perfect freedom of movement for various pressures. The annular cavity between *4* and *5* serves as a steam jacket and permits *4* to expand and contract freely.

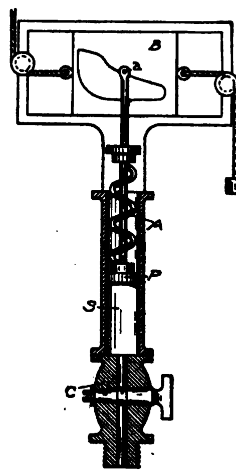


Fig. 1. Original Watt Indicator

Piston *8* is made of a good quality of steel and is hardened to prevent its surface from wearing. It is ½ square inch in area. Small grooves around its outer surface provide a steam packing, and the moisture and oil which collect in these grooves prevent too much leakage of steam past the piston. At the center of the piston is a boss or hub which projects both upward and downward. The upper part of the hub is threaded inside to receive piston rod *10*. The upper edge of this hub is so formed that it fits nicely into a circular groove in the bottom side of the nut of the piston rod. The hub also has a slot cut diametrically across it, which permits the flat portion of the spring with head to fit on a curved bearing on the piston screw *9*. When making connection between piston *8* and piston rod *10*, it is very essential that the hub shank fit tightly against the bottom of

the circular groove in the bottom of the shoulder of the piston rod. If this connection is correctly made, a perfect alignment of the piston is assured.

The swivel head *11* is threaded at the bottom, so that it can be screwed into the piston rod. By so doing the height of the pencil and, therefore, the atmospheric line can be raised or lowered as desired.

Cap *2* is an important part of the indicator as it holds all the moving parts in place and guides the piston. It has two projections

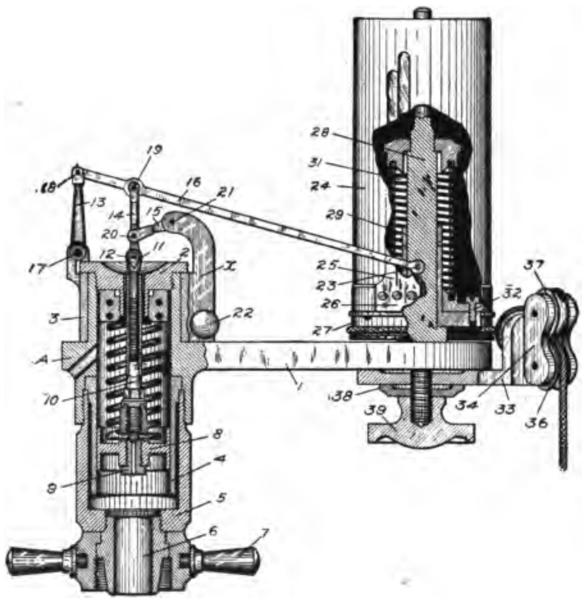


Fig. 2. Part Section of Crosby Indicator

of different diameter on the lower side. The projection with the larger diameter is threaded so that the cap can be screwed into the cylinder. The smaller projection is also threaded to engage with like threads on the spring head which holds it firmly in position. Cap *2* holds sleeve *3* in position in a recess formed for the purpose. This sleeve carries the pencil mechanism, parts of which are *15*, *13*, etc. The arm *X*, which carries lever *15* of the pencil mechanism, is made integral with the sleeve. A handle *22* is provided by which

the pencil point is brought in contact with the paper. This handle is threaded and, being in contact with a stop screw on plate *1*, permits a very delicate adjustment of the pencil point to the surface of the paper on the drum. It is desired to have the pressure just great enough (but no greater) to secure a visible diagram on the paper.

The pencil mechanism, consisting of links *13*, *14*, *15*, and *16*, is a very important part of the indicator. Its essential kinematic principle is that of a pantograph. This mechanism must be so designed and adjusted that the path of pencil *23* is at all times parallel to the path of piston *8*. The links are so proportioned that the movement of the pencil is six times that of the piston.

The indicator card is held on paper drum *24*—which is made of very light metal—by means of clips *25* and *26*. Drum *24* fits on a base *27*, which carries a spring *31* on a central projection *28*; this spring brings the drum back to its initial position when the indicator is detached from the moving parts of the engine or when a return stroke is made.

The direction in which the cord may be conducted from the drum can be adjusted by means of guide pulleys *36* and *37*, which are attached to the indicator by nut *39* and frame *33*.

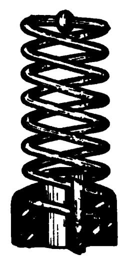


Fig. 3. Indicator Piston Spring

The piston spring, Fig. 3, which should be of a good quality of spring steel, must be carefully made and tested, and also carefully handled, as the accuracy of the results depends in a large measure upon the accuracy of the spring.

It will be noted in Fig. 2 that the piston or pressure spring is placed within cylinder *4*, when the indicator is put together for use. This brings the spring in contact with the live steam and, as a consequence, errors may be recorded due to the uneven heating of the spring and contained parts. To eliminate the possible inaccuracies due to heat, the manufacturers have constructed indicators with the spring on the outside, as illustrated in Fig. 4. Aside from the elimination of errors by the use of the outside spring, convenience is obtained in that the spring is more accessible and may be removed or changed without taking the indicator apart, which can not be done with the spring inside. Furthermore, the spring does not get

very much warmer than the surrounding atmosphere, so it is not necessary to allow the indicator to cool before removing it.

Fig. 4 also shows how an indicator may be easily changed from an ordinary steam indicator to an indicator suitable for gas engine work. The change is made by reducing the size of the cylinder to ¼ square inch in area and increasing the strength and weight of the pencil mechanism. By these changes, an indicator may be used on gas engines with good results.

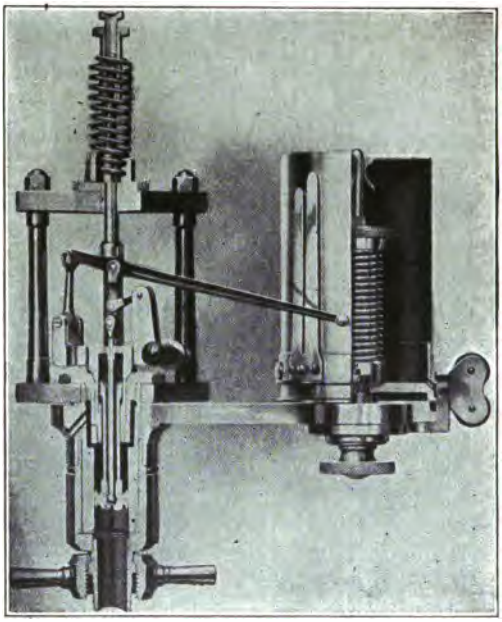


Fig. 4. Crosby Indicator with Outside Spring

The Crosby indicator is ordinarily made with a drum 1½ inches in diameter, this size being suitable for high-speed work. If, however, a larger diagram is desired and the speed is low, a drum 2 inches in diameter can be furnished.

Tabor Indicator. The Tabor indicator, Fig. 5, with outside spring, reducing motion, and electrical attachment, is similar in construction, operation, and essential characteristics to the Crosby and other indicators, though there are details of design for which the respective makers claim advantage over other makes. One feature

of the Tabor which is essentially different from the Crosby and the Thompson is its pencil mechanism. As was shown in Fig. 2, the Crosby pencil mechanism consists of a system of levers, which gives

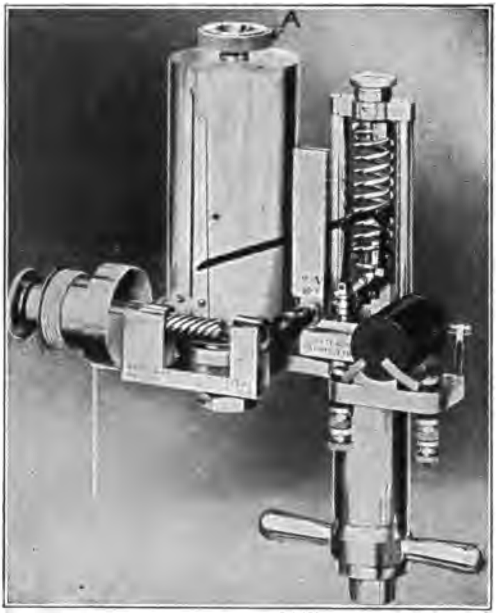


Fig. 5. Tabor Indicator with Outside Spring

to the pencil a straight-line motion parallel with that of the piston, with possible slight errors, especially on high cards. As shown in Fig. 6, the scheme for obtaining the parallel or straight-line movement of the Tabor indicator is different from that of the Crosby. It has the connecting links corresponding to *13*, *14*, and *16* in the Crosby, Fig. 2, but in place of link *15*, there is substituted another arrangement. A stationary plate *1*, with a curved slot *2*, is fastened in an upright position to the cap. On the pencil bar is a roller bearing *3*, which is secured to the bar by a pin. This roller moves freely in the curved slot in the guide and controls the motion of the pencil bar. The position of the slot and the guide upright is so adjusted

Fig. 6. Tabor Device for Straight-Line Movement

and the guide roller is so placed on the pencil bar that the curve of the guide slot controls the pencil motion and absolutely compensates for the tendency of the pencil to move in a curve. There is a minimum of friction in this movement and guide, and no disturbance from inertia has been detected by the most careful tests.

American Thompson Indicator. The American Thompson indicator, Fig. 7, does not differ greatly in general appearance from

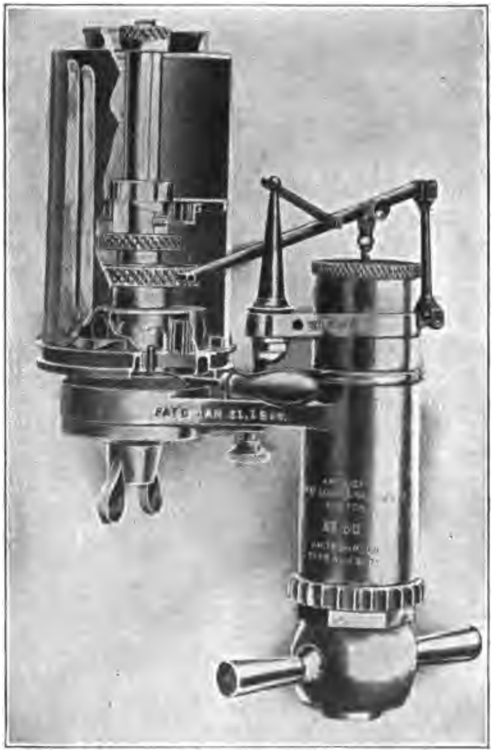


Fig. 7. American Thompson Indicator

other indicators, but a close comparison will show some difference in the details of construction. For instance, it is evident that the arrangement of the levers, which make up the pencil motion, differ slightly from that of the Crosby indicator. Each maker, of course, makes the assumption that his particular arrangement is the best. In most cases, the purchaser must use his own judgment in the matter. Again the connection of the spring to the piston is different on the

TABLE I

Constants of Indicator Springs

Scale of Springs	Maximum Safe Pressures to Which Springs Can Be Subjected — Pounds Pressure per Square Inch with ½ Square Inch Area Piston	
	To 200 Revolutions per Minute	To 300 Revolutions per Minute
8	10	6
10	15	10
12	20	15
16	28	22
20	40	32
24	48	40
30	70	58
32	75	62
40	95	80
48	112	95
50	120	100
60	140	115
64	152	125
80	180	145
100	200	160
120	240	195
150	290	250
200	375	330

Thompson than on others, in that the spring screws directly into an enlargement on the upper side of the piston, thus being rigidly attached to the piston as well as to the pencil motion at the top instead of having a semi-flexible connection by means of a ball and socket joint, as in the Crosby. These two points are the distinguishing ones of the Thompson indicator. The construction of its cylinders, piston, paper drum, etc., are about the same as for those previously described. In the figure, a portion of the drum is shown cut away, disclosing the detent motion, the operation and purpose of which will be described later.

The piston of an indicator is usually .798 inch in diameter, which is equivalent to ½ square inch area. This size piston with springs is designed to indicate pressures up to 250 pounds. When higher

pressures than 250 pounds are used, a piston .564 inch in diameter, representing an area of $\frac{1}{4}$ square inch, is substituted for the $\frac{1}{2}$-inch piston. This doubles the capacity of the spring and makes it possible to indicate up to 500 pounds.

Since it is the capacity of the spring that limits the height of the indicator card and since the various springs are made to resist a definite amount of pressure, it is necessary that the proper capacity of spring be used at all times. This capacity is designated by the term "scale of spring" which means the amount of pressure required on the piston per square inch of area to raise the pencil point 1 inch. For example, if one hundred pounds steam pressure is being used and a spring having a scale of 40 is placed on the indicator, the height of the resulting card will be $100 \div 40 = 2\frac{1}{2}$ inches. The capacity of the spring is always marked upon it, as 40, 60, etc. The manufacturers of the Tabor indicator recommend the use of springs having capacities for various conditions of speed and pressure as given in Table I.

If an engine is running at a speed not exceeding two hundred revolutions per minute and the steam pressure being used is 180 pounds, the scale of spring to be used is 80. If the revolutions per minute (r. p. m.) be increased to between two hundred and three hundred, then a spring of 80 pounds should not be used for pressures higher than 145 pounds. This table is about the same as that given by other makers of indicators. A common rule for determining the capacity of the spring to be used is to multiply the scale of the spring by $2\frac{1}{2}$ and subtract 15, the result being the limit of pressure to which the spring should be subjected. To illustrate: Assume a spring having a scale of 60. Then $60 \times 2\frac{1}{2} = 150$. $150 - 15 = 135$. Therefore, 135 pounds pressure is the ultimate capacity of a 60-pound spring, which approximately checks Table I.

From the foregoing discussion of the indicator and the study of its construction, it is evident that the essentials of a good indicator are summed up by Professor Thurston in the following paragraphs:

(1) Such form and construction as will insure its meeting the prescribed general conditions—accuracy of representation of variations of steam pressure and simultaneous movement of the piston at all times.

(2) Such simplicity of form as will make it free from liability to accident and failure in operation.

(3) Such lightness of parts and rigidity of whole, as will prevent any inaccuracies of indications arising from its inertia.

(4) It should be easily, conveniently, and safely attachable and unmovable and handily manipulated.

(5) Stiffness, lightness, and exactness of standardization are prime essentials. Springs should be exactly standard. Moving parts as light as consistent with proper strength and stiffness; stationary parts should be carefully proportioned and rigid.

INDICATOR SPRING TESTING

Apparatus. As the accuracy of the action of the indicator spring is of primary importance in obtaining correct indications,

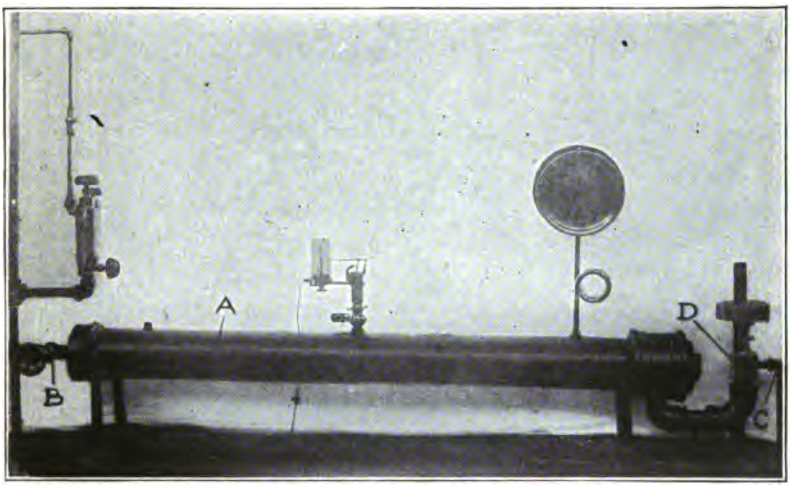


Fig. 8. Spring Testing Device

some means must be employed for testing indicator springs. A very simple but efficient apparatus for testing indicator springs is shown in Fig. 8, which consists of a drum *A*, made of 4- or 5-inch extra heavy pipe having steam-tight joints. Steam is admitted to the drum at *B*, and permitted to pass out at *C*, through a piston regulating valve *D*, carrying a disk and weights.

The indicator and a standard test gauge for checking are attached in the manner shown. The pressure regulating valve *D* is very sensitive and responds to a very slight change of pressure. By placing the desired weight on the disk and adjusting valve *B*, the pressure of steam in the cylinder is maintained at a constant value. If

the pressure should rise higher than desired, the piston valve *D* rises, permitting the escape of steam through pipe *C*, and in this way maintains a constant pressure in the drum.

Spring Calibration. To test or calibrate the spring, proceed as follows: Put the indicator together properly and see that the

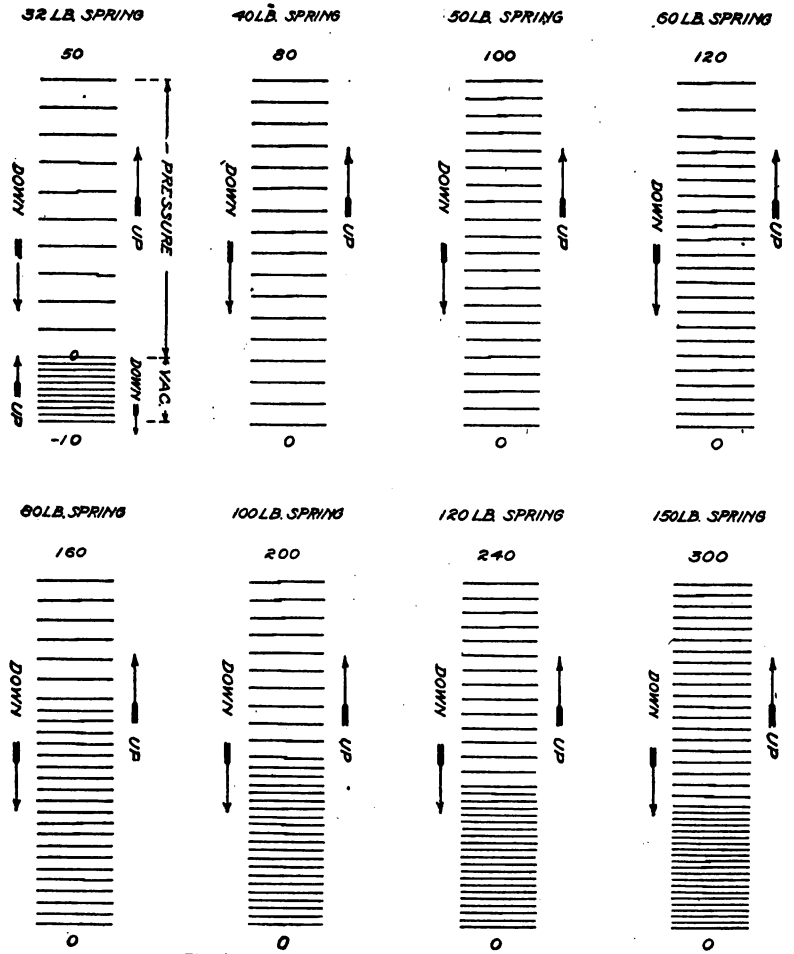


Fig. 9. Cards Showing Records of Spring Tests

piston is oiled and in place. Attach the indicator in the usual manner. After the indicator has been warmed up by permitting steam to act on it, put the desired weight on the disk and spin it, at the same time moving the indicator drum by pulling the cord and hold-

ing the pencil against the paper drum, thus recording the pressure on the paper. Proceed in this manner by equal increments of pressure until the capacity of the spring has been reached, then reduce the pressure by the same increments until zero is reached. The operation of taking both the upward and the downward readings should be continuous, stopping only long enough to change the weights and to make the proper indications.

Measuring the Cards. After the cards have been taken, they may be measured by means of a scale in the usual manner. The pressures thus measured should check within a fairly close margin of the readings corresponding to the gauge and the tester.

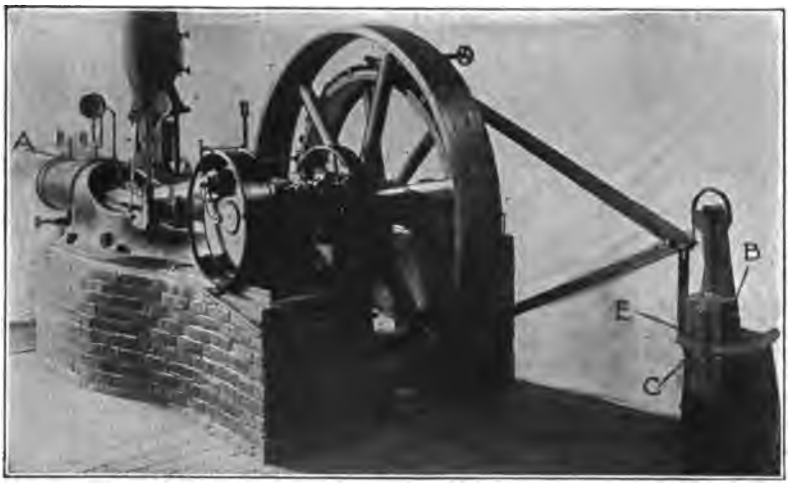


Fig. 10. Engine Showing Two Indicators Screwed Into Cylinder

The cards, Fig. 9, show the records obtained from tests of various springs. It is evident that some of the records taken with increasing pressures do not coincide with the corresponding record when going down or with decreasing pressures. For very accurate work, the spring should be used with the piston and in the indicator with which it was tested.

Engine Connection. The attachment of the indicator to the engine should be such that the pressure of the steam on the indicator piston is exactly the same as that acting at the same instant on the engine piston. In order to secure this result, the steam connection between the indicator and the engine should be amply large and direct.

If possible, the indicator connection should be screwed directly into the cylinder as shown at A, Fig. 10. In making this attachment, a hole is drilled in the cylinder and a connection is made to the indicator by means of a standard $\frac{1}{2}$-inch pipe and a proper valve or cock. The hole in the cylinder should be drilled in the clearance space, where the piston will at no time cover any portion of the opening, and where no strong currents of steam will sweep directly into the passage.

If it is possible to remove the cylinder heads, it should be done before drilling, so as to properly locate the holes and to remove any chips which may happen to fall into the cylinder while it is being

Fig. 11. Method of Attaching Indicator to Locomotive

drilled. If it is not feasible to remove the cylinder heads, the cylinders should be carefully blown out with steam before running the engine, as much damage may result from the chips in the cylinder The indicators should be attached in an upright position if possible. It is best to have an indicator attached to each end of the cylinder, so that cards may be taken simultaneously from both ends. Before drilling the holes, a general plan or scheme should be studied out for the attachment of the indicator and its necessary appliances, as the type of engine (whether vertical or horizontal), the type of crosshead, and the necessary room for operation may be quite different for each case; so a strict rule can not be laid down for this procedure.

Suffice it to say that generally the indicator can be attached to the side of the cylinder or to the top, as shown in Fig. 10. Figs. 10 and 11 illustrate the methods used in attaching indicators to a simple engine and a locomotive, respectively. It will be observed that all of these connections are short and direct, that the indicators are in an upright position, and that the cord of the indicator is led straight to the crosshead connection.

Sometimes it is not convenient to use two indicators or it may be that the engineer does not care to bear the cost of two, so only one is used. When only one is used, a pipe leading from each end of the cylinder is connected to the indicator by means of a three-way cock, as

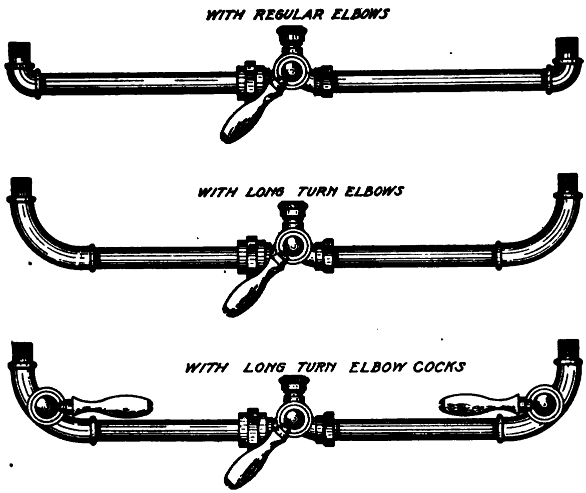


Fig. 12. Typical Three-Way Valve

shown in Fig. 12. By the use of this cock, the indicator is put in connection first with the head end (h. e.) and then the crank end (c. e.) of the engine. This should be done with as little loss of time as possible so that the cards will represent, as nearly as possible, actual conditions of pressure in the cylinder. The three-way cock, or any other cock which may be used in the system, should have an opening as large as that in the cylinder connection of the indicator. It should also have a hole in one side for the purpose of freeing the indicator and its connections from any water that may come over with the steam.

Avoid Long Pipe Connection. Long connections between the indicator and the engine cylinder should be avoided in all cases.

Experiments conducted at Purdue University have demonstrated the fact that any pipe connection between the indicator and the engine is likely to affect the action of the indicator. Under ordi-

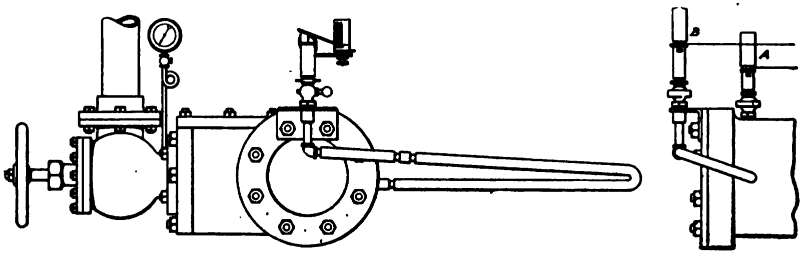


Fig. 13. Method of Attaching Indicator Piping to Engine Cylinder

nary pressures and speeds, a length of pipe over 3 feet in length so distorts the card that the results obtained are useless except for approximate work.

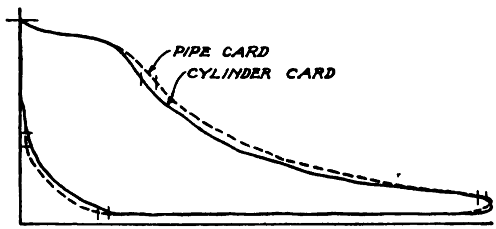


Fig. 14. Indicator Card Showing Effect of Changing Connecting Pipe

In order to determine the effect of long pipe connections between the indicator and engine, upon the form of the cards, a series of tests

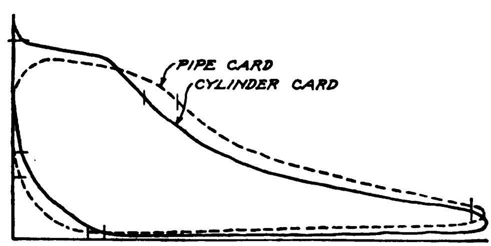


Fig. 15. Indicator Card Showing Effect of Changing Connecting Pipe

were conducted in the Engineering Laboratory of Purdue University under the direction of Dean W. F. M. Goss. The results of these experiments formed the basis of a paper which Dean Goss

presented before a meeting of the A. S. M. E. at St. Louis in May, 1896. The experiments were made in connection with a $7\frac{3}{4}$- by 15-inch Buckeye engine. Very great care was taken in the selec-

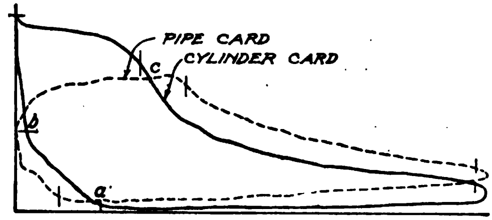


Fig. 16. Indicator Card Showing Effect of Changing Connecting Pipe

tion and testing of the indicators and in their manipulation, in order to insure that any distortion which might occur in the cards would be due entirely to the pipe connections. The indicators were

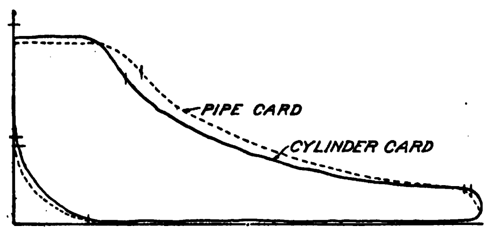


Fig. 17. Indicator Card Showing Effect of Changing Connecting Pipe

attached to the cylinder, as shown in Fig. 13, both being connected at the same end so that the indicator pistons would be exposed as nearly as possible to identical conditions.

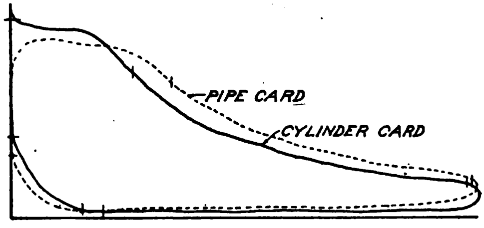


Fig. 18. Indicator Card Showing Effect of Changing Connecting Pipe

The indicator *A* and the cards obtained therefrom will be hereinafter designated as cylinder indicator and cylinder cards and it is

assumed that this indicator will give indications true to the conditions in the cylinder.

The indicator *B* and the cards obtained therefrom will be designated as the pipe indicator and pipe cards, respectively, and it is assumed that any perceptible difference in the cards obtained from

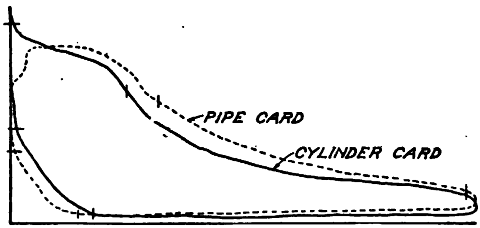


Fig. 19. Indicator Card Showing Effect of Changing Connecting Pipe

the cylinder indicator and from the pipe indicator will be due entirely to the pipe connections.

Pipe connections of 5, 10, and 15 feet were used, the length of pipe being measured from outside of the cylinder walls to the end of the coupling under the indicator cock. Care was taken in securing easy bends in the pipe so as not to retard the action of the steam.

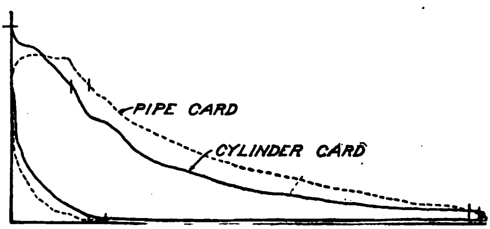


Fig. 20. Indicator Card Showing Effect of Changing Connecting Pipe

The pipes were also properly insulated in order to avoid in so far as possible any condensation.

The method of procedure was to run the engine for a short length of time, until the desired speed, cut-off, pressures, etc., were obtained, then cards were taken simultaneously from the two indicators. Two cards were taken from each indicator, then the indicators were interchanged and two more cards taken from each, thus obtaining four cylinder cards and four pipe cards.

Figs. 14 to 22 inclusive illustrate the effect of the pipe on the form of the indicator card with the engine running under various

conditions. A study of these cards reveals the fact that the length of the piping to the indicator affects very materially the area of the diagram. The events of the stroke, while remaining unchanged, are apparently generally made later by the use of the long pipe, but in some cases, some of them are earlier.

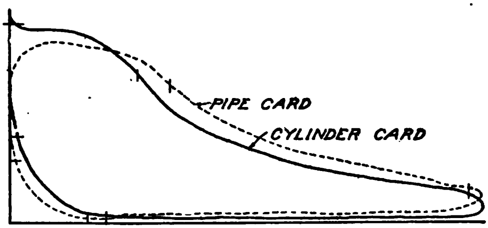


Fig. 21. Indicator Card Showing Effect of Changing Connecting Pipe

It is hoped that what has been said is sufficient to point out the importance of the short indicator connection.

Continuous Diagrams. From the study of the indicator, it has been obvious that the indicator card gives an indication of what is taking place in the cylinder at a specific moment. This being the case, it is practically impossible to obtain by its aid determinations

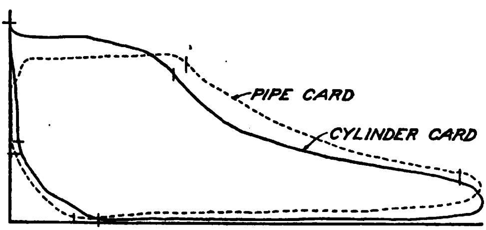


Fig. 22. Indicator Card Showing Effect of Changing Connecting Pipe

that are to be relied upon when the engine is working under constantly varying conditions, as in gas engines, locomotives, marine engines, and rolling-mill engines. To meet this demand, an attachment has been developed whereby it is possible to take a continuous card, thus getting exact determinations under the most variable conditions.

Crosby Device. Fig. 23 represents a Crosby indicator equipped for taking continuous diagrams. The special drum is designed so as to be applied to any Crosby indicator, and uses a roll of paper

2 inches wide and 12 feet long upon which the series of diagrams are traced. The roll of paper is located within an opening in the drum. From the roll, the paper passes around the outside of the

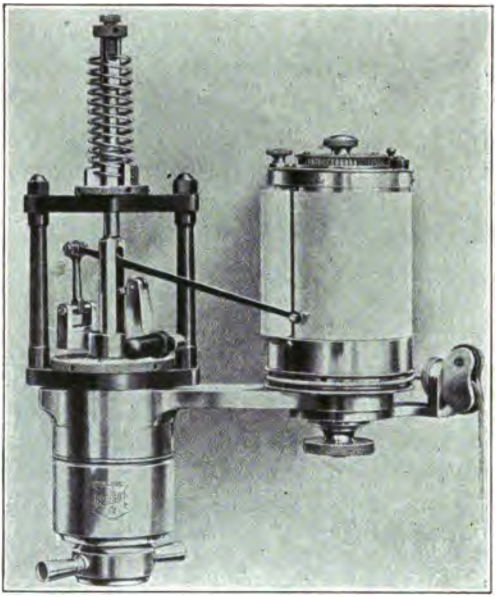


Fig. 23. Crosby Indicator with Continuous Diagram Device

drum, thence inward to a central cylinder to which it is attached. In taking cards the paper rolls up on the central cylinder, which is concentric with the drum, and may be withdrawn through the top and easily detached. On the top of the drum is a knurled head,

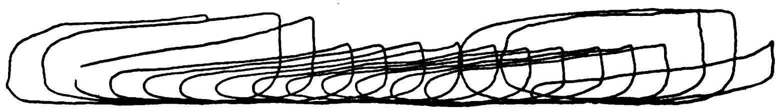


Fig. 24. Continuous Diagram from Rolling-Mill Engine

loosely attached to the drum spindle, which controls the distance between diagrams. Adjustment can be made so that from 6 to 100 diagrams can be made per foot of paper.

Fig. 24 illustrates a series of continuous cards taken from a rolling-mill engine and clearly shows the widely varying conditions.

Fig. 25 shows cards taken from an automatic cut-off rolling-mill engine.

After providing proper means for attaching the indicator to the cylinder, the next important step is to provide a convenient and at the same time correct reducing or drum motion.

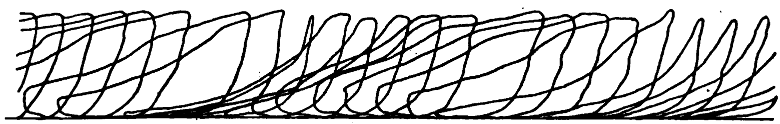


Fig. 25. Continuous Diagram from Automatic Cut-Off Rolling-Mill Engine

Reducing Motions. In the description of the indicator, it was noted that the indicator card is held on the circumference of the paper drum by means of clips. Since the circumference of the drum is much less than the length of stroke of the engine, some means must be provided to reduce the motion of the drum. As each engine and its location will be different, no strict rule can be given whereby one can at once provide a reducing motion, but each case must be studied and the best means possible provided to meet the exigency. A few examples and principles will be given and doubtless they will suggest others to meet specific cases.

Brumbo Pulley. The Brumbo pulley, Fig. 26, is easily and quickly made and can be attached to almost any engine. If it is to be used for only a short time, it may be constructed of wood, care being exercised in having close fitting joints. If the engine is to be indicated frequently, it is better to make the reducing motion of metal in order that the wear in the joints may be minimized. The reducing lever A is pivoted overhead to some temporary support or, if it is to be permanently attached to the engine, some permanent support, such as an upright post or bracket, may be attached to the frame of the engine, as in Fig. 10. The segment S is made fast to the lever A, so that its semicircumference is true with its pivot

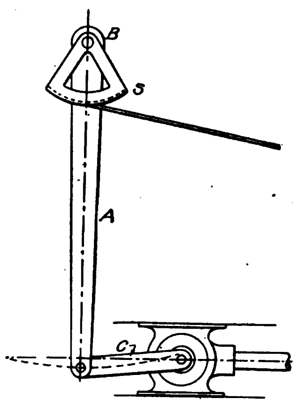


Fig. 26. Brumbo Pulley Reducing Motion

point *B*, upon which the lever swings. The sector may be set at any angle with the lever. The lower end of the lever *A* is attached to the crosshead through the link *C*. The length of the lever *A* should be at least one and one-half times the length of stroke of the engine. The length of the connection *C* may be about one-half of the length of stroke, but it may be greater. When the crosshead is at its mid-position, the lever *A* should be vertical. During the stroke of the engine, the link *C* should swing equally above and below a horizontal position.

With this form of reducing motion, the cords may be led in any direction in a vertical plane from the sector and one or more cords may be led off to different indicators. The face of the sector should be true and the radius must have a value sufficient to give the required motion to the drum. To fulfill this condition, the radius of the sector must bear the same relation to the length of the lever *A* as the proposed length of the indicator diagram bears to the stroke of the engine.

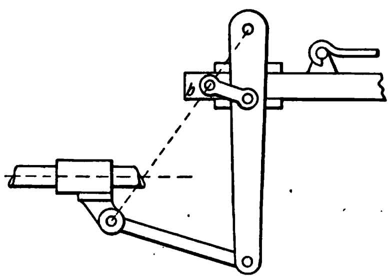


Fig. 27. Diagram Showing Principle of Brumbo Pulley

Example. Suppose it is desired to make a reducing motion for a 10- by 16-inch engine. Assume the length of the lever *L* to be 24 inches and the required length of card *D* to be 4 inches. The length of stroke *S* is 16 inches. Designating the radius of the sector as *R*, then

$$R : L :: D : S$$

Solving for *R* in the above engine

$$R : 24 :: 4 : 16$$
$$16\ R = 96$$
$$R = 6 \text{ inches}$$

The principal objection to the Brumbo pulley is that it is not interchangeable, that a different one is required for engines of different types and sizes. If it is carefully made and attached, however, it will give results with very slight inaccuracies. The design illustrated in Fig. 27 is theoretically correct, and its construction and operation are so clearly shown that a detailed description is not deemed necessary. It has been successfully used for experimental work in colleges and universities for a number of years.

Pantograph. Another form of reducing motion, known as the pantograph, is shown in Fig. 28. It is placed horizontally, with the pivot *B* resting on a support opposite the crosshead when in mid-position. The pivot *A* is attached to the crosshead, usually by having the stud *A* inserted in a hole drilled in the crosshead. If the pivot *B* is adjusted to the proper height and at the right distance from the crosshead, the cord from the indicator may be attached to the pin *E* without any pulleys, which is very desirable. The length of the diagram is adjusted as desired by means of the movable piece *C D* and the pin *E*. The pin *E* must always be on a line joining the pivot points *A* and *B*. The pantograph gives correct results when in good condition and properly attached but, on account of the large number of joints, it may become unsatisfactory. This type is usually used on engines having a long stroke and where it is not convenient to attach a Brumbo pulley or its equivalent. It is applicable to all types of engines of any length of stroke. In two of the three forms of reducing motions just described, there are chances for inaccuracies. There is an error in the Brumbo pulley which may or may not be very large, depending on the proportions of the levers, and there may be lost motion in the many joints of the pantograph. The inaccuracy of the Brumbo pulley can be materially decreased by using, instead of the sector, a *sliding bar*, as in Fig. 11. This sliding bar is supported by means of brackets and is attached to the lever by means of a pin which works in a slot similar to that at the lower end of the lever. As the crosshead moves to and fro, the sliding bar likewise moves, and its motion is proportional to that of the crosshead at all points of the stroke. All things being considered, the principle of the reducing motion shown in Fig. 11 is all that could be desired; it is especially suited for locomotive engines.

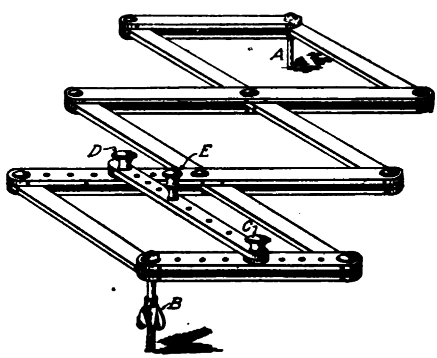


Fig. 28. Pantograph Device for Reducing Motion

Reducing Wheel. Nearly all makers of indicators manufacture

a reducing wheel apparatus which serves the same purpose as the lever and pantograph types of reducing motions. A type of this apparatus is shown in Fig. 29; also it may be seen in Fig. 5 attached to the indicator. The following description is given by the makers.

The reducing wheel is composed of a supporting base piece *A*, provided with short standards *B* that form bearings for the worm shaft on which the flanged pulley *D* is rotated, the outer bearing being a pivot which receives the entire thrust of the shaft, thus reducing the friction to a minimum. It is connected directly to the indicator upon the projecting arm that supports the paper drum, and the teeth of the worm shaft mesh directly into the teeth on the drum carriage *C*. Connected with the base piece is a spring case *E*, and on the extreme end of the worm shaft is secured a collar *F* through which freely slides a clutch pin, one end of which is securely fastened to a thumb piece for operating it.

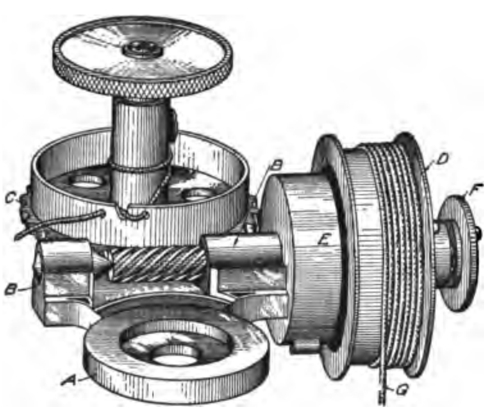


Fig. 29. Tabor Reducing Wheel

The flanged pulley *D* runs freely and independently on the worm shaft, and has on its outside a clutch-shaped hub. To this pulley is connected the actuating cord *G*, which should encircle it a sufficient number of times to have its length, when unwound, a little more than equal the length of the stroke of the engine. The other end of the cord is secured to the crosshead of the engine or to a standard bolted thereto or to any moving part that has an exact similar motion, and must be connected in line from the pulley.

Enclosed in the spring case *E* is a small, plain, spiral steel spring which operates solely to return the pulley to its starting point, after it has been revolved in one direction by the forward movement of the crosshead. As this pulley has an independent rotating back-and-forth motion on the worm shaft, the necessity of unhooking the cord when the indicator is not being operated is entirely overcome.

The paper drum is rotated forward by means of the pulley through its worm shaft, engaging with the worm gear on the paper drum carriage, and is rotated in the opposite direction by the action of its own retracting spring. On top of the paper drum is a knurled thumb piece (see *A*, Fig. 5) made with a projecting pin on its under side to engage with a similar pin located in the top of the drum; this is to be used by the operator in moving the paper drum slightly forward, preparatory to taking a diagram, in order to prevent it from striking against its stop on the return motion.

To operate this device, first, select a pulley whose diameter is about one-twelfth of the length of the engine stroke in inches. Properly place this pulley upon the worm shaft by removing the clutch and then sliding the pulley onto the shaft, being particular that the small hole in the pulley brass disks sets over the projecting pin in the cover of the spring case. Then replace the clutch by pushing it onto the shaft as far as it will go, and secure it there by means of the set screw.

Now place the indicator on the engine in such a position that the side of the pulley *D* will be parallel with the motion of the crosshead. Run out the loose end of the cord to a distance of at least 12 or 18 inches beyond the extreme forward travel of the crosshead, still leaving a turn or two of the cord on the pulley unwound. While holding the cord, allow it to gradually recede and rewind itself on the pulley until its loose end has reached a point coincident with the extreme backward travel of the crosshead. If only a slight tension of the cord exists at this point, it will be sufficient, and the cord may then be attached to the selected point on the crosshead. The cord tension may always be adjusted either by winding the cord on, or unwinding it from, the pulley, as the case requires, one increasing and the other decreasing the tension.

A much lighter cord can be used in proportion as the sizes of the pulleys increase.

When the crosshead, with cord connected, is at its extreme forward travel, there should be just sufficient tension on the spring enclosed in the spring case to take up all slack of the cord when running, without overtaxing the spring. If, upon starting the engine, the cord should at first run unevenly on the pulley, turn the indicator to one side slightly until a perfect and uniform winding of

the cord is obtained, which can always be easily secured. When the pulley is running, motion to the paper drum is obtained by pushing in the swivel collar to which the clutch pin is secured.

When ready to take diagrams, after placing the paper on the drum it is necessary first to advance the drum away from its stop fully ¼ inch, which can be done by turning with one hand the knurled top thumb piece. While holding drum in this position, with the other hand push in gently the swivel collar to start the paper drum in motion. The motion of the paper drum can at any time be stopped for removing diagrams taken and renewing the paper by withdrawing the swivel collar or by turning the top thumb piece, the latter method being preferable, as it prevents damage from too severe contact with the paper drum stop. The stopping of the paper drum will not affect the motion of the pulley, which will continue to revolve independently while the engine is in motion until the cord is disconnected.

With the indicator are usually furnished three different size pulleys. Unless otherwise specified, pulleys furnished are 1, 2, and 3½ inches in diameter. These pulleys are sufficient for the average work required of an indicator. Larger sizes can be obtained if needed.

The reducing wheel form of reducing motion has many points of advantage over the pantograph and lever in that it is conveniently attached and it allows the operator to start and stop the motion of the paper drum without disconnecting the cord where attached to the crosshead, which is an annoying thing to do under some circumstances. The reducing wheel does not work under high speeds as satisfactorily as for the lower speeds, which is, perhaps, one of its most objectionable features. When the indicator is kept in constant use for several hours at a time, some trouble may be experienced with the cord becoming worn and breaking during a test.

From this study of the reducing motion, it is evident that much must be left to the discretion of the operator as to the selection and attachment of the motion.

Crosby Reducing Wheel. The Crosby reducing wheel, shown in Fig. 30, is attached directly to the cylinder cock of the steam engine, and has connected to it the steam engine indicator which it is to serve; thus it forms a base of support for the latter, and receives all the strains and shocks in the operation of the engine, to

the relief of the indicator. Its bearings are made comparatively frictionless by the introduction of minute balls running in hardened tool steel races, thus affording lightness and freedom of movement. The cord pulley is horizontal, to allow the cord leading to the engine crosshead to take any direction the circumstances may require, without regard to the position of the indicator.

Whenever the reducing wheel is to be attached to a vertical engine, an elbow nipple is provided, which will allow the cord pulley

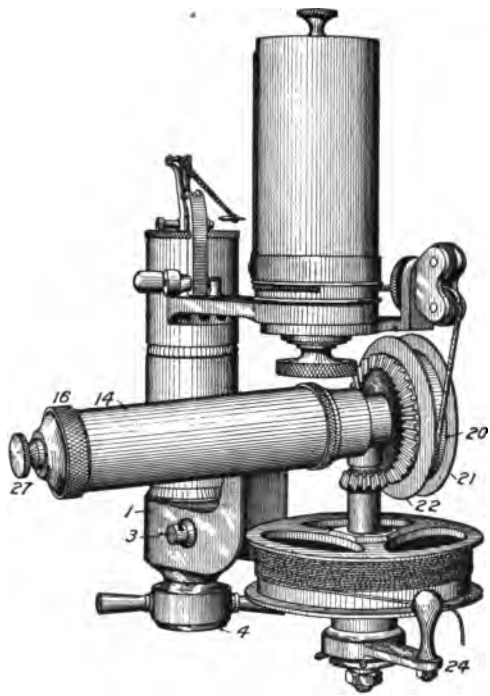


Fig. 30. Crosby Reducing Wheel Attached to Indicator

to travel in the proper plane for guiding it to the crosshead of the engine with the indicator in an upright position as usual.

OPERATION RULES

The Crosby reducing wheel is attached directly to the cylinder cock of the steam engine by means of union *4* of standard *1*. Connect the indicator to standard *1* with the paper drum standing over spring *14* and the indicator guide pulley in a proper position over stroke pulley *20*.

To attach the cord guide: Loosen cord guide *24* by means of the screw beneath the cord pulley; then move it around to the proper position for the cord to pass directly through the hole in the cord guide without rubbing, to the crosshead of the engine and tighten it in place.

To take up the tension spring: Release thumb screw *27* in the end of the shaft within spring tube *14;* withdraw knurled spring head *16* from its square end, and turn it one or more squares as may be desired.

To adjust the stroke pulley: Remove knurled disk *21*, which holds in place stroke pulley *20* on the gear shaft; place on the shaft the stroke pulley desired; replace the disk and screw it up firmly.

To attach the indicator cord: Wrap the indicator cord one or more turns around stroke pulley *20*, passing the end through the hole in, and around, the hook of knurled disk *21*.

When used with other indicators, loosen bolt *3* in the side of standard *1*, where it is attached to the cylinder cock of the steam engine; remove the bushing and insert another fitted to the indicator to be used.

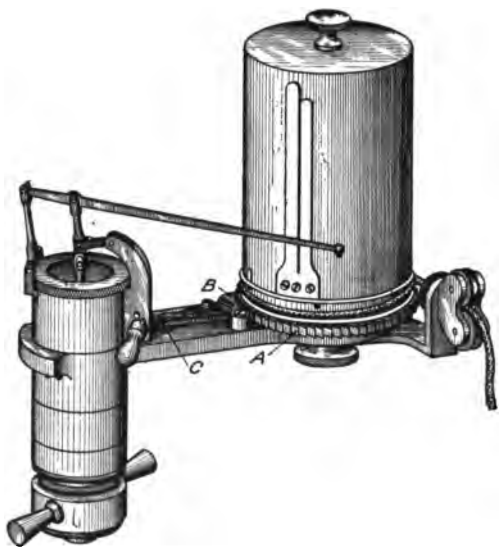


Fig. 31. Indicator with Detent Attachment

Simultaneous Indicator Cards. In making complete and reliable tests of power plants, it is desirable that all of the cylinders of the compound and multiple engines be taken simultaneously at a given signal. This requirement, if the indicators are hand-operated, would necessitate an operator at each cylinder, which would be expensive and besides would not insure the simultaneous taking of the cards. The makers of indicators have met this need by supplying the market with an electrical attachment *B*, Fig. 5, which is attached to each indicator that is to be used. It is not thought necessary to give a description of the construction and operation of the appliance, but suffice it to say that by pressing an electric button, the pencils of all the indicators in the circuit are simultaneously brought in contact with the paper and thus a record is made.

Detent Attachment. Another attachment that can be obtained and which is of much convenience to the operator at times is the detent attachment, shown in Fig. 31. It consists of a ratchet *B* that fits into the teeth of the wheel *A*. When the operator wishes to stop the motion of the paper drum, he pushes the lever *C*, which causes *B* to come in contact with *A*, as shown. Thus, the operator may change the card, and do other things without disconnecting the indicator cord from its crosshead connection. This is a very desirable attachment when indicating high-speed engines and when taking cards on a locomotive on the road where conditions are not ideal for using the indicator.

Fig. 32, illustrates the detent attachment used on the Thompson improved indicator. With this new improved detent motion, in order to stop the paper drum it is only necessary to move lever *A* in the direction traveled by the paper drum until the drum releases itself. The lever must then be returned to its original position. When ready to take the diagram, turn forward the paper drum, by means of the milled rim *B* on top, until it catches, causing the drum to revolve in the usual manner; then take the diagram and release the drum as described above. Before taking the diagram, see that the parts are cleaned and well oiled. To oil, remove the knurled nut *F*, take off the paper drum, then with the wire clip (which is sent with each indicator) remove the auxiliary spring case *H* by catching the end of the clip in the notches of the spring case, turning it forward until it releases from the catches; then move the spring and inner sleeve *I*. After cleaning and oiling, replace the inner sleeve *I* by inserting it into the drum so that the pin on the outside of the sleeve will enter the slot inside of drum bearing, and turn it until it comes to a stop; then with the wire clip catch hold of the auxiliary spring holder *H* and give the auxiliary spring *E* a tension of about one-fourth turn, and catch the points on the spring case *H* into the slots provided for them.

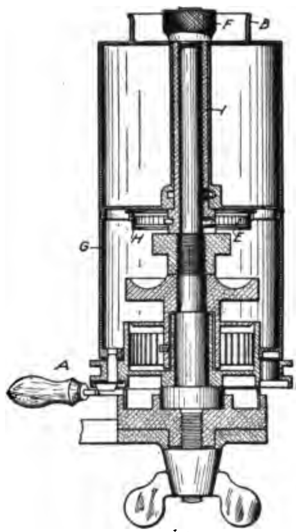


Fig. 32. Thompson Detent Attachment

ASSEMBLING AND ADJUSTING THE INDICATOR

Thus far, it has been the endeavor to mention the chief features of different makes of indicators and to point out the important points to be observed in the attachment of the indicator for obtaining a correct movement of the paper drum. Before proper results can be obtained, however, it is absolutely necessary that the indicator be properly put together. By way of illustration, reference will be made to Fig. 2, which shows the Crosby indicator with all of the parts connected together ready for connection to the engine cylinder.

Assembling Crosby Indicator. When the indicator is removed from the engine cylinder, the spring should in every case be disconnected from the piston and well cleaned before putting away. To remove the piston and spring, unscrew cap *2;* then take hold of sleeve *3* and lift all the connected parts from the cylinder. This gives access to all the parts for the oiling and cleaning, which should be thoroughly done after each time the indicator is used. None of the pins as *17*, *19*, etc., should be removed except in making repairs and they should be kept well oiled in order to reduce friction. After removing the spring and cleaning the indicator thoroughly, connect the parts together, leaving out the spring, and put the indicator away. It is evident that the indicator must be put together each time it is used and in doing this great care must be exercised in order to insure satisfactory working of the parts. It is important to notice that on the under side of the shoulder of the piston rod *B*, Fig. 33, there is a circular channel formed to receive the upper edge of the slotted socket of the piston *A*.

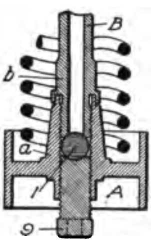


Fig. 33. Placing Indicator Spring

Connecting Piston Rod. Whenever it is desirable to connect the piston rod with the piston, either in the process of attaching a spring, or for the purpose of testing the freedom of movement of the piston in the cylinder without a spring, *be sure to screw the piston rod into the socket as far as it will go;* that is, until the upper end of the socket *a* is brought firmly against the bottom of the channel *b* in the piston rod. This insures a perfectly central alignment of the parts and, therefore, a perfect freedom of movement of the piston within the cylinder.

Attaching Spring. To attach the spring, place the piston rod *B*, Fig. 33, in a hollow wrench provided for that purpose, so that the wrench will encircle the hexagonal part of the piston rod. Holding the hollow wrench with the piston rod in place, in a vertical position, place the spring over the wrench so that round head *1* will be in the concave portion of the end of the piston rod. Unscrew set screw *9* until it is almost removed from the piston. Invert the piston and insert the transverse wire at the lower end of the spring in the slot in the socket of the piston. Screw the piston on to the piston rod as far as it will go. Remove the wrench, and holding sleeve *3* and cap *2* (see Fig. 2) in an upright position, so that the pencil lever will drop to its lowest position, engage the threads of swivel head *11* with those inside the piston rod, and screw it up until the threads on the lower projection of cap *2* engage those in the spring head. Continue the process until the spring is screwed up tightly on cap *2*. After this, holding sleeve *3* in one hand, with the other turn the piston swivel onto the piston rod. It sometimes occurs that when the piston rod is screwed up on the swivel, the atmosphere line (the line inscribed by the pencil on the paper drum) is not properly located, so it becomes necessary to unscrew the piston rod until the atmosphere line is at the proper height. A little practice will soon teach one about how the parts should be left in order to bring the atmosphere line in the correct position. It is important to avoid having the atmosphere line too high, as trouble might result if the pencil point moved above the top of the paper drum.

Having thus secured the spring to the piston and cap, take the open wrench and turn set screw *9* snugly against the head on the spring. It is important that this should not be done until the spring has been securely fastened to the piston and to cap *2*, for there is then less likelihood of error in alignment. After completing the successive steps named, oil the piston with a good cylinder oil, insert it in the cylinder, screw cap *2* down tightly, which will cause all of the parts to assume their proper places.

Testing Action. Before placing the spring on the indicator, it is well to test the indicator in order to determine whether or not it is in good condition. To do this, put the indicator together carefully, omitting the spring, oil the piston, and place the parts in the cylinder.

Then raise the pencil as high as it will go and release it. If it returns to the bottom of its own accord, it is an assurance that everything is in alignment and that there is little friction in the moving parts.

Adjustment. *Length of Indicator Cord.* Having carefully connected the parts of the indicator and attached it to the engine cylinder in the proper manner, the next step is to adjust the length of the indicator cord, which should be as short as possible. If it is impossible to use a short cord, then a fine steel or brass wire should be used. The builders of indicators usually furnish a braided cord which has been well stretched and which gives good results. Sometimes it is convenient to make a loop in the end of the cord, which is engaged by a small hook attached to the reducing motion. One method for adjusting the length of the cord is as follows: The hook *A*, Fig. 34, is attached to the indicator cord. The cord *B* from the

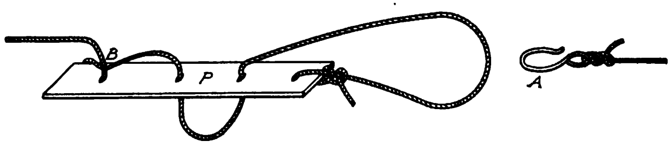


Fig. 34. Device for Altering Length of Indicator Cord

reducing motion which passes through the holes in the plate *P*, as shown, is adjusted in length by slacking it at the point *B* and slipping the plate along the cord. To avoid an accident, in the way of injuring the indicator or the reducing motion, it is best to determine as nearly as possible the necessary length of the cord before hooking up to the reducing motion. To determine the length of the cord, take hold of the end of it, and the hook to which it is to be attached; holding them in their relative positions, follow the motion of the reducing lever, keeping the cord tight, thus pulling the drum from one stop to the other, observing if the string must be lengthened or shortened to insure the drum traveling the proper distance. Having determined the length of the cord, hook the two cords together and ascertain whether or not the indicator drum strikes the stop at either end of the stroke.

Adjusting Card and Pencil. Having made the proper adjustment of the length of cord, put the indicator card on the paper drum, being sure that it is smooth and even, as otherwise the diagram will

not be a true representation of the pressure in the cylinder. Considerable practice is required before one can put cards on the drum smoothly and rapidly, but this is desirable. After having caught both ends of the card by the clips, bend the ends over so that they will not interfere with the pencil arm.

Adjust the pencil stop so that the pencil can bear only very lightly on the paper when in the proper position. Always use a pencil with a smooth sharp point, so the lines obtained will be plain and fine. A fine pointed pencil produces less friction when in contact with the paper, which is desirable. The pencil may be easily sharpened by using a small piece of sandpaper or a file.

TAKING CARDS

Everything being in readiness, attach the cord to the reducing motion and, with the pencil off of the card, open the cock and let steam pass into the cylinder of the indicator for a few strokes to warm it up; then put the pencil in contact with the paper for one revolution, after which turn the cock so that no steam is in the cylinder. Again hold the pencil point in contact with the paper, thus getting the atmosphere line. The atmosphere line should always be taken last, in order that there is assurance that all the parts of the indicator are of the same temperature.

Condition of Indicator. It is important to know that the indicator is working properly at the beginning of the test, so after taking the first card, the indicator may be tried to see if it is working correctly. Open the cock and let the piston make a few strokes, close the cock, place the pencil in contact with the paper, at the same time turning the drum by hand; if the pencil retraces the original atmosphere line, or if after a slight pressure up or down on the piston, the pencil returns to the atmosphere line, it is evidence of its being in good condition. If the indicator fails to do the above, the pencil movement is not free in its joints, there is lost motion, or the piston does not move freely in the cylinder of the indicator.

A card should be taken from both ends of the cylinder, for there can be no positive assurance that the same conditions exist in both ends. In fact, oftentimes there is quite a difference between the cards for the head end (h. e.) and the crank end (c. e.), due to inaccuracies of the valve motion and other defects.

Sample Indicator Card. The cards are made of a good grade of white paper, one side being finished smooth. It is on the smooth side that the diagram is made. Manufacturers frequently furnish blank indicator cards having printed on the back of each a set of blank spaces to fill out, which is convenient for filing for the purpose of preserving important data. All of the blank spaces may not be filled out each time, but the more important points should never be neglected. The following form has been used by different successful engineers:

Date....................	Name of operator.............	Builder of engine................
Time....................	Owner of engine.........	Kind of valve motion..........
Diam. of cylinder........	Kind of work done by engine................	Kind of steam valves...........
Length stroke...........		Kind of exhaust valves.........
R.P.M..................	Remarks................	Kind of condensers............
Speed of piston..........		Kind of heater................
Diam. piston rod........		Kind of boiler ....
Area steam port.........	Barometer reading..............	Kind of fuel..................
Area exhaust port........		Temperature feed water............
Piston clearance.........		Temperature hot well..............
Port clearance...........		Water per hour.................
Boiler pressure...........		Coal per hour.................
Initial pressure..........		
M.E.P..................		

Indicator Card Analysis. *Meaning of Lines of Diagram.* Fig. 35 illustrates a typical indicator card with all reference lines and events of the stroke designated by letters. It is to be remembered that the indicator card shows the relation between piston position and pressures in the cylinder. So on the diagram, all vertical lines or ordinates represent pressures and all the horizontal lines or abscissas represent distances. Bearing this distinction in mind, the pressure in the engine cylinder at any piston position, measured along the horizontal line, can be obtained by measuring the vertical height of the diagram at the point representing the piston position. There

has been a set of lines placed on the indicator card shown, known as reference lines. These lines are *O Y*, *Y K*, and *O X*. The other lines *DE*, *EF*, *F G*, *GH*, etc., are drawn by the indicator, as is also the atmosphere line *AB*, and it is the result of one indication from one side of the engine piston, say the head end side. The diagram for the crank end would be quite similar, but reversed in position on the paper.

The reference lines are the atmosphere line *A B*, boiler pressure line *Y K*, clearance line *O Y*, and the vacuum or absolute pressure line *O X*. The atmosphere line is drawn by the indicator, when both sides of the indicator piston are acted upon by the pressure of the atmosphere only. Since this line is used as a reference line in meas-

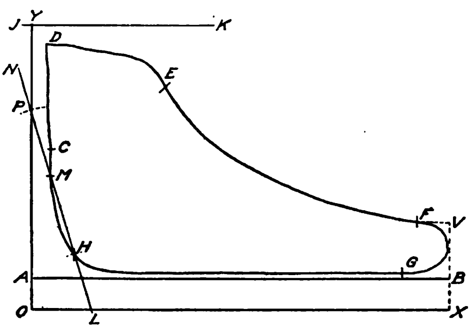


Fig. 35. Typical Indicator Card

uring all pressures, it should be carefully located. The vacuum or absolute pressure line *O X*, the zero line of pressure, is drawn by hand below and parallel to the atmosphere line a distance by scale equal to the barometric pressure, which at sea level is 14.7 pounds per square inch.

The line of boiler pressure *J K* is drawn above and parallel to the atmosphere line a distance by scale equal to the boiler pressure by gauge. The distance between the boiler pressure line and the line *D E* represents the loss in pressure that occurs while the steam is passing from the boiler to the cylinder of the engine.

The clearance line *O Y* is drawn at right angles to the atmosphere line and at a distance from the extremity of the diagram equal to the per cent of clearance of the engine multiplied by the horizontal length of the diagram. That is, if the clearance is 2 per cent of

the piston displacement and the length of the card is 4 inches, then the distance between the extremities of the card and the clearance line would be 2 per cent of 4, which is .08 of an inch. By the term "clearance" is meant the volumetric space between the piston and the bottom of the valve when the engine is on dead center and this may differ in amount at each end of the cylinder. Obviously this space must be filled with steam from the boiler at the initial pressure at every stroke of the piston. It is, therefore, desirable to have as small a per cent of clearance as is consistent with good design, in order to eliminate the loss of live steam. For slow-speed engines the clearance space needs to be small—about 2 to 5 per cent, whereas for high-speed engines, it may run as high as 10 per cent.

Measurement of Clearance. If the per cent of clearance is not given by the builders, it becomes necessary to measure it if any scientific study is to be made of the performance of the machine. Professor John E. Sweet gives a very simple plan for obtaining the per cent of clearance as follows:

> See that the piston and valves are made tight, and the valves disconnected. Arrange to fill the clearance space with water through the indicator holes, or through holes drilled for the purpose. Turn the engine on dead center; make marks on the crosshead and guides; weigh a pail of water, and from it fill the clearance space. Weigh the remaining water so as to determine how much is used. Then weigh out exactly the same amount of water (as is used), turn the engine off the center, pour in the second charge of water, and turn the engine back till the water comes to the same point that it did in the first instance. Make another mark on the crosshead and guide, and the distance between these marks is exactly what you really wish to know; that is, it is just what piston travel equals the clearance. If it takes one pound of water to fill this space and to admit another pound, the piston must be moved 1 inch; then the clearance bears the same relation to the capacity of the cylinder as 1 inch bears to the stroke of the piston. Thus, under these circumstances, in an engine of 10-inch stroke, it would be said to have 10 per cent clearance.

The above method would be correct when the engine is new, the piston and valves being tight and there being no leaks while the trial is being made; on the other hand, if the engine is old and the piston and valves are worn, it would only give approximate results. In such a case, it would be advisable to ascertain the per cent of clearance from the card by the following simple method. In Fig. 35 draw the straight line LN from a point L on the vacuum line in

such direction that it will cut the compression curve at two points, as H and M. Now with a pair of dividers, set one leg on the point L and adjust the other to the point H. With the dividers thus set, place one leg in the point M, where LN cuts the compression curve, then sweep an arc cutting LN at P. Erect a line perpendicular to the vacuum line and passing through the point P, which establishes the clearance line $OAPY$.

Events of the Cycle. While the steam engine is making one complete revolution, four separate and distinct events occur, namely, the admission, cut-off, release, and compression of steam. The point in the stroke where these events occur can be very accurately determined from the indicator diagram. Corresponding to the above events, there are six distinct lines on the card, namely, admission, steam, expansion, release, compression, and back pressure. By properly analyzing the diagram in Fig. 35, these events and lines are easily determined.

The admission line CD shows the rise of pressure in the cylinder due to the opening of the steam valve permitting steam to enter the cylinder. The point C indicates the point in the stroke at which the admission of steam took place.

Steam line DE is drawn while the valve remains open and steam is being admitted to the cylinder.

At the point E, the valve closes the steam port, thus cutting off the supply of steam to the cylinder, hence E is the point of cut-off (c. o.).

As the motion continues, the volume back of the piston is increased and the pressure drops, due to the expansion of the steam, giving the expansion curve EF.

The point of release occurs at F, where the valve uncovers the exhaust port, permitting the steam to escape from the cylinder.

As soon as the point of release F is reached, the pressure begins to drop and by the time the point G has been reached on the return stroke practically all of the steam has been exhausted, hence FG is called the release line.

The back pressure line GH indicates the amount of pressure against the engine piston while making the return stroke. On noncondensing engines, it is either coincident or above the atmosphere line. On condensing engines, it is found below the atmosphere line a distance depending upon the amount of vacuum being maintained.

At the point H the exhaust port is closed and the steam remaining in the cylinder is compressed. The line HC indicates the rise of pressure due to the compression.

The events of the stroke—as cut-off at E, release at F, compression at H, and admission at C—are not always easily located on

the card, as there may not be a distinct change in the curve where these different events occur. The engineer must use his best judgment in locating these points. As an aid in making a decision, if one carefully inspects the diagram, it will be noted that when cut-off takes place at *E*, for instance, the steam line is concave and the expansion line convex downward, hence where these two opposite curves join must be the point at which cut-off occurs. Even this suggestion does not always hold but it will serve as an aid.

The events are usually expressed in per cents of the stroke. To obtain these per cents, proceed as follows: Locate the events as described above. Consider the point of cut-off as located on the card shown in Fig. 36. From this point, draw a line perpendicular to the atmosphere line, cutting it at the point *G*. Measure the length

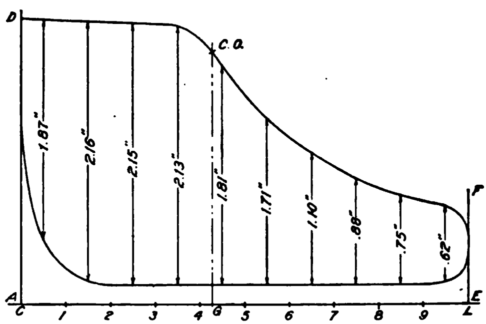


Fig. 36. Indicator Card Laid Out for Determining the M. E. P.

of the card between the ordinates *C D* and *F L;* measure also the distance *C G*. The per cent of stroke at which cut-off occurs would be $\frac{C\,G}{C\,L} \times 100$. *C G* equals 1.62 inches; *C L* equals 3.72 inches. Therefore

$$\text{Per cent of cut-off} = \frac{1.62}{3.72} \times 100 = 43.5 \text{ per cent}$$

To find the per cent of release, admission, or compression, proceed in the same manner as for cut-off, always measuring the distances from the admission end of the card.

Pressures. In discussing an indicator card, five different pressures are frequently considered, namely, initial, terminal, and back pressure, pressure at the events, and the mean effective pressure

The initial pressure is the pressure in the cylinder at or near the beginning of the stroke. It would be measured on a perpendicular line from the atmosphere line to the steam line at D in Fig. 35.

The terminal pressure is the pressure measured above the vacuum line at the end of the stroke. It is the pressure that would have been acting against the piston at the end of the stroke if the steam had not been released earlier. It is measured by extending the expansion curve until it cuts a perpendicular at the end of the card at V, Fig. 35. VX measured to scale gives the amount of the terminal pressure.

Back pressure, which is the pressure the piston works against on the return stroke, is the distance between the atmosphere line AB and the back pressure line GH, Fig. 35.

The pressure at the events is obtained by scaling a perpendicular line drawn from the points in question to the atmosphere line.

The mean effective pressure, usually written m.e.p., is the net average pressure that acts on the piston throughout the entire stroke. It is evident from examination of Fig. 35 that the m.e.p. is the average height of the card multiplied by the scale of the spring used in the indicator.

There are two general ways of obtaining the m.e.p., viz, by the use of a planimeter and by the ordinate method.

In Fig. 36, CL is the atmosphere line and CD and FL are perpendiculars drawn at the end of the card. Divide CL into ten equal divisions, as *1*, *2*, *3*, etc., and midway between *C-1*, *1-2*, etc., draw the lines shown. Measure these lines and mark their lengths on them. When this is done, obtain the sum of all of these lengths, which, in this case, is 15.18 inches; and 15.18 inches divided by 10 gives 1.518 inches, the average height of the card. If the scale of the spring used was 40 pounds, then 1.518 inches multiplied by 40 gives 60.72 pounds as the m.e.p.

The ordinate method for finding the area of a card, then, is to divide the atmosphere line into ten or more equal divisions and, half way between these divisions, erect ordinates and divide the sum of all the ordinates by the number of lines and multiply by the scale of the spring.

The average m.e.p. for one revolution would be the average of the two mean effective pressures as determined from cards taken from both the head and the crank end of the cylinder.

The number of divisions into which the card is divided could have been twenty as well as ten or any other number, but as ten or twenty makes the computations simple, they are usually taken. More accurate results will be obtained if a greater number of divisions are made, other things being equal.

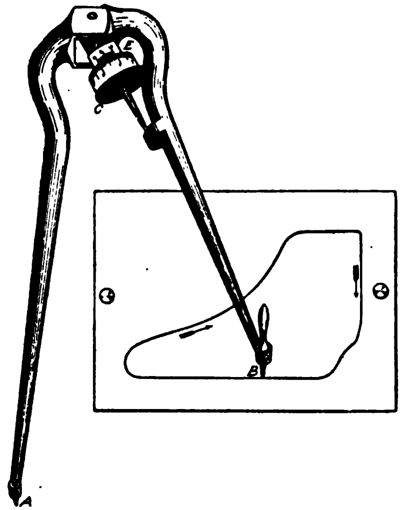


Fig. 37. Measuring Area of Diagram by Means of Planimeter

Determinaton of Mean Effective Pressure by Planimeter. A more accurate way of obtaining the m.e.p. is by using an instrument called the planimeter, of which there are several types in common use. The Amsler polar planimeter is one of the most simple, and as shown in Fig. 37 is about one-half the size of the instrument. It consists of two arms free to move about a pivot and a roller graduated in square inches and tenths of square inches. A vernier is placed with the roller so the areas may be read in hundredths of a square inch. The point *A* is kept stationary and the tracer *B* is moved once around the outline of the diagram. The area in square inches of the diagram is read from the roller *C* and the vernier *E*.

INSTRUCTIONS FOR USE OF PLANIMETER

The diagram should be fastened to some flat unglazed surface, such as a drawing board, by means of thumb tacks, springs, or pins. The point *A* is pressed into the paper so that it will hold in place; *B* is set at any point in the outline of the diagram; and the roller is set at zero. Follow the outline of the diagram carefully in the direction of the hands of a watch, as indicated by the arrows in Fig. 37, until the tracer has moved completely around the diagram. The result is then read to hundredths of an inch from the roller. Suppose, after tracing over the outline, we find that the largest figure that has passed the zero of the vernier is 3; the number of graduations (tenths) that have passed the zero are 5; and the graduation (hundredths) on the vernier that exactly coincides with a graduation on the roller is 9. Then the area is 3.59 square inches.

Often at the start the roller is not adjusted so that the zeros coincide, but the reading is taken and subtracted from the final reading. Thus, if the

first reading is 4.63, and the second 7.31, the area is 7.31 − 4.63 or, 2.68 square inches. In case the second reading is less than the first, add 10 to the second reading, then subtract.

Briefly stated, to find the m.e.p. of a diagram, first ascertain the area of the card by use of the planimeter, multiply the area obtained by the scale of the spring, and divide the product by the length of the card in inches. This may be demonstrated in the following manner: If in a rectangle A equals area in square inches, L equals length in inches, and H equals height in inches, then

$$A = LH$$

If P is the average pressure, then

$$H = \frac{\text{average pressure}}{\text{scale of spring}}$$

or

$$H = \frac{P}{\text{scale}}$$

Substituting this value in the equation $A = L\ H$, we have

$$A = \frac{L \times P}{\text{scale}}$$

or

$$P = \frac{A \times \text{scale}}{L}$$

The planimeter is a very valuable instrument to an engineer in taking indicator cards, the results obtained being very accurate. Ten or twelve diagrams can be measured by this instrument in the same time that is necessary to measure a single card by the method of ordinates.

It is well to run over the area two or three times and take an average, as the tracing of the diagram can not be absolutely correct at any time.

PHYSICAL THEORY

In the study of any subject, there are always a number of technical terms that need to be clearly understood before a proper understanding of the subject is obtained. This is especially true with the steam engine, and the indicator, and with steam which is the motive force of each. It is, therefore, thought best to treat these terms and a study of the properties of steam together, at this point, in order to be able to go more deeply into the study of the uses of the indicator.

Pressure. Pressure is the force tending to compress a body and is usually expressed either in pounds per square inch, pounds per square foot, inches of mercury, or inches of water.

Boiler Pressure. Boiler pressure is the pressure of steam in pounds per square inch above atmosphere as indicated by a steam gauge.

Absolute Pressure. Absolute pressure is the pressure as obtained from absolute vacuum. At the sea level, the pressure of the atmosphere is usually taken as 14.7 pounds per square inch; hence, if at sea level, a steam gauge reads one hundred pounds, the absolute pressure would be $100+14.7$ or 114.7 pounds per square inch absolute.

Atmospheric Pressure. Atmospheric pressure is the pressure the atmosphere exerts upon a body. It is usually measured in inches of mercury, as obtained from a barometer, hence it is sometimes spoken of as barometer pressure. Since the weight of 1 cubic inch of mercury is known to be .49 pounds, the reading of a barometer can be easily converted into pounds per square inch. If a barometer reads 28 inches of mercury, the pressure of the atmosphere expressed in pounds per square inch would be $28 \times .49 = 13.72$.

Pressure below atmosphere is also given in inches of mercury and sometimes in inches of water. If it is desired to obtain the absolute pressure in pounds per square inch, reduce the reading in inches of mercury to pounds per square inch and subtract this amount from the atmospheric pressure expressed in pounds per square inch.

EXAMPLE. A vacuum gauge on a steam engine condenser reads 26.1 inches of mercury. The barometer stands at 29.12 inches of mercury. What is the absolute pressure in the condenser in pounds per square inch?

SOLUTION. 29.12 inches of mercury is equivalent to $29.12 \times .49 = 14.268$ pounds per square inch and 26.1 inches is equivalent to $26.1 \times .49 = 12.789$ pounds per square inch. Therefore, the absolute pressure in pounds per square inch in the condenser is $14.268 - 12.789 = 1.479$.

Work. The unit of work is called a *foot pound* and it is equal in amount to the energy required to lift one pound one foot high. It is to be noted that it is the product of a force times the distance through which it acts. If a weight of 50 pounds be raised 7 feet high, then 50×7 or 350 foot pounds of work would be expended.

Heat. *Temperature.* By temperature is meant simply the thermal condition of a body with reference to its capability of transferring heat to other bodies. If two bodies are placed in contact and the first gives more heat to the second than it receives, we say that No. 1 is hotter than No. 2. If the first receives more heat than it gives, No. 2 is hotter than No. 1. If both bodies give and receive the same amount of heat, they are of the same temperature.

According to our theory, it is evident that temperature depends upon the energy of molecular vibration. If the temperature rises, it means that the molecular vibration, and consequently the energy increases; if the temperature falls, the energy of molecular vibration decreases. Evidently, a point must finally be reached when this energy of vibration is zero and the molecules are at rest. At this temperature, there is no heat and we call it the absolute zero. This zero is evidently much below the zero of the ordinary scale.

Thermometers. In order to determine just how hot a body is, we must compare its temperature with that of some substance whose degree of heat we know. As it would be impossible to keep several bodies at different degrees of heat for comparison, we must resort to some other means. A simple method is to use some substance whose volume changes a definite amount for a definite change in temperature and always has the same volume for the same temperature. Mercury and alcohol are suitable substances and may be placed in a glass bulb, to which is connected a glass tube of small bore. All the air is drawn out of the tube, and the end is sealed so that the thermometric substance can expand or contract in a vacuum. The tube having been sealed, the bulb is placed in melting ice and the height of the mercury in the tube noted. It is then placed in steam (or boiling water) at atmospheric pressure and the height of the column again noted. On the Fahrenheit scale, the melting point is called 32 degrees and the boiling point 212 degrees, and the intervening space is divided into 180 equal parts. In the centigrade scale, the melting point is called zero degree and the boiling point 100 degrees; there are 100 equal intervals between them. Thus we see that 180°F. = 100°C., or 1°C. = 1.8°F.

EXAMPLE. What is the temperature of 50°C. on the Fahrenheit scale?

$$50°C. = 50 \times 1.8 = 90°F. \text{ above the melting point}$$
$$= 90 + 32 = 122°F.$$

In order to compare temperatures, we place the thermometer in contact with the substance whose degree of heat we wish to know and then observe the height of the liquid column in the thermometer. The height of this column depends upon the expansion of the thermometric substance and indicates the intensity of heat, or the temperature as we commonly call it. We use a thermometer to measure the *intensity* of heat, but not the *quantity* of heat.

Unit of Heat Quantity. For measuring the intensity of heat, the degree is the unit; for measuring the quantity of heat, we have another unit, which is the amount of heat necessary to raise one pound of water from 61° F. to 62° F. This is called the British thermal unit (B.T.U.). To raise one pound of water from 60° F. to 62° F., or to raise two pounds from 60° F. to 61° F., will require 2 B.T.U.

One B.T.U. is equivalent to 778 foot pounds of work. If one pound of coal liberates 12,000 B.T.U. when burned, it is capable of producing $12{,}000 \times 778 = 9{,}336{,}000$ foot pounds of work.

The above value of the B.T.U. in foot pounds of work is known as the mechanical equivalent of heat, that is, 778 foot pounds.

Horsepower. Horsepower is the arbitrary standard used for measuring the power of a steam engine. It is said to have been originally established by James Watt from experiments conducted with dray horses on the streets of London. Its value is, however, considerably above that of the ordinary horse. It is defined as being equal to lifting 33,000 pounds one foot high in one minute of time. It will be noted that the horsepower takes into account the following factors: *force, distance,* and *time.* This being true, it is desirable to have an expression in the form of an equation to express the horsepower of an engine. The common formula for steam engines is

$$\text{h.p.} = \frac{PLAN}{33000}$$

in which P equals the mean effective pressure in pounds per square inch; L equals length of stroke in feet; A equals area of cylinder in square inches; N equals number of revolutions per minute (r.p.m.); and 33,000 equals equivalent foot pounds of work per minute in one horsepower.

Analyzing the equation, it is found that it conforms to the defini-

tion of work previously given. For instance, A, the area of the piston in square inches, times P, the mean effective pressure in pounds per square inch, is equal to the total force on the piston which acts through the stroke a distance of L feet. Hence, the expression PLA gives the foot pounds of work done during one stroke. In the definition of horsepower, it was noted that the time element was considered, so we have $(PLA)\times N$ divided by 33,000, fulfilling the definition of a horsepower, since N involves the element of time.

Indicated Horsepower. The indicated horsepower (i.h.p.) is the computed horsepower of an engine as obtained by using an indicator diagram taken from the engine cylinder. From this diagram is determined P, the mean effective pressure, which is substituted in the equation just given.

EXAMPLE. Given an engine having a cylinder 10 inches in diameter and a stroke 16 inches in length, running at 180 r.p.m. The mean effective pressure on the piston as obtained from the indicator card is 75 pounds on both the head and the crank end of the cylinders. Required the horsepower of the engine.

SOLUTION. In this example P equals 75 pounds; L equals $16\div 12$, or 1.33 feet; A equals πR^2 equals 3.1416×5^2 or 78.54 square inches; and N equals 180 r.p.m.

Substituting these values in the formula

$$\text{h.p.} = \frac{PLAN}{33000}$$

we have

$$\text{h.p.} = \frac{75\times 1.33\times 78.54\times 180}{33000} = 42.75$$

This is for one end of the cylinder only. For both ends, we get the total

$$\text{h.p.} = 42.75\times 2 = 85.5 \text{ (approximately)}$$

ENGINE CONSTANT. It is evident from the foregoing problem that, for a given engine, some factors in the h.p. formula remain constant. These constants are: the area of the piston, the length of the stroke, and the abstract number 33,000. It is convenient when making a large number of computations to determine what is known as the engine constant, a factor which saves considerable time and reduces the chances of error. Since the area of the piston on the crank end is smaller than that on the head end by an amount equal to the area of the piston rod, the engine constant for the crank end is always slightly smaller than for the head end.

EXAMPLE. Find the constant for both h.e. and c.e. of the engine in the preceding problem, whose piston rod was 1¾ inches in diameter. The engine constant for the head end is

$$C_{h.e.} = \frac{L\,A}{33000}$$

in which L equals $16 \div 12$, or 1.33 feet, A equals 3.1416×5^2 or 78.54 square inches.

$$\therefore \qquad C_{h.e.} = \frac{1.33 \times 78.54}{33000} = .00316$$

For the crank end, the area of piston is reduced by the area of the 1¾-inch piston rod, which area is equal to 2.40 square inches. The effective area for the crank end is, therefore, $78.54 - 2.40$, or 76.14 square inches.

$$\therefore \qquad C_{c.e.} = \frac{1.33 \times 76.14}{33000} = .003068$$

Having obtained the engine constant, in order to obtain the indicated horsepower (i.h.p.) it is only necessary to multiply the m.e.p. on the piston for each end of the cylinder by the engine constant for that end and by the number of revolutions.

In Table II, the approximate i.h.p. of an engine is easily found by multiplying the constant, corresponding to the diameter of the piston, by the piston speed and by the m.e.p. Or, in other words, the constants in the table equal the h.p. for an engine with a given diameter of piston having a piston speed of one foot per minute and a m.e.p. of one pound. The piston speed of any engine is equal to the length of stroke in feet multiplied by twice the number of revolutions. For instance, in the 10- by 16-inch engine already referred to, the piston speed in feet per minute would be 1.35 feet $\times$ $180 \times 2 = 478.8$ feet per minute.

If the diameter of the piston is an even number, the constant is found in the second column; if it contains a fraction, the constant is found by following the column horizontally until the required fraction is reached. The constant multiplied by the piston speed in feet per minute and by the m.e.p. in pounds per square inch gives the i.h.p. approximately.

EXAMPLE. An engine runs at 75 r.p.m. and the stroke is 4 feet. If the m.e.p. is 48 and the piston is 27⅝ inches in diameter, determine the i.h.p.

SOLUTION. From Table II, the constant for a piston 27⅝ inches in diameter is .0178355. The piston speed is $4 \times 75 \times 2 = 600$ feet per minute. Then

$$\begin{aligned} \text{i.h.p.} &= .0178355 \times 600 \times 48 \\ &= 513.66 \text{ (approx.)} \end{aligned}$$

TABLE II
Engine Constants

Diameter of Cylinder	Even Inches	+⅛" or .125	+¼" or .25	+⅜" or .375	+½" or .5	+⅝" or .625	+¾" or .75	+⅞" or .875
1	.0000238	.0000301	.0000372	.0000450	.0000535	.0000628	.0000729	.0000837
2	.0000952	.0001074	.0001205	.0001342	.0001487	.0001640	.0001800	.0001967
3	.0002142	.0002324	.0002514	.0002711	.0002915	.0003127	.0003347	.0003574
4	.0003808	.0004050	.0004299	.0004554	.0004819	.0005091	.0005370	.0005656
5	.0005950	.0006251	.0006560	.0006876	.0007199	.0007530	.0007869	.0008215
6	.0008568	.0008929	.0009297	.0009672	.0010055	.0010445	.0010844	.0011249
7	.0011662	.0012082	.0012510	.0012944	.0013387	.0013837	.0014295	.0014759
8	.0015232	.0015711	.0016198	.0016693	.0017195	.0017705	.0018222	.0018746
9	.0019278	.0019817	.0020363	.0020916	.0021479	.0022048	.0022625	.0023209
10	.0023800	.0024398	.0025004	.0025618	.0026239	.0026867	.0027502	.0028147
11	.0028798	.0029456	.0030121	.0030794	.0031475	.0032163	.0032859	.0033561
12	.0034272	.0034990	.0035714	.0036447	.0037187	.0037934	.0038690	.0039452
13	.0040222	.0040999	.0041783	.0042576	.0043375	.0044182	.0044997	.0045819
14	.0046648	0047484	.0048328	.0049181	.0050039	.0050906	.0051780	.0052661
15	.0053550	0054446	.0055349	0056261	.0057179	.0058105	.0059039	.0059979
16	.0060928	.0061884	0062847	.0063817	.0064795	.0065780	.0066774	.0067774
17	.0068782	.0069797	.0070819	0071850	.0072887	.0073932	.0074985	0076044
18	.0077112	0078187	0079268	.0080360	.0081452	.0082560	.0083672	.0084791
19	.0085918	0087052	.0088193	0089343	.0090499	.0091663	0092835	.0094013
20	.0095200	0096393	.0097594	.0098803	.0100019	.0101243	.0102474	.0103712
21	0104958	0106211	0107472	.0108739	.0110015	.0111299	.0112589	.0113886
22	.0115192	.0116505	.0117825	.0119152	.0120487	.0121830	.0123179	.0124537
23	.0125902	0127274	.0128654	.0130040	.0131435	.0132837	.0134247	.0135664
24	.0137088	.0138519	.0139959	.0141405	.0142859	.0144321	.0145789	.0147266
25	.0148750	.0150241	0151739	.0153246	.0154759	.0156280	.0157809	.0159345
26	.0160888	.0162439	.0163997	.0165563	.0167135	.0168716	.0170304	.0171899
27	.0173502	0175112	.0176729	.0178355	.0179988	.0181627	.0183275	.0184929
28	.0186592	.0188262	.0189939	.0191624	.0193316	.0195015	.0196722	.0198436
29	.0200158	.0201887	.0203624	.0205368	.0207119	.0208879	.0210645	.0212418
30	.0214200	.0215988	.0217785	.0219588	.0221399	.0223218	.0225044	.0226877
31	.0228718	.0230566	.0232422	.0234285	.0236155	.0238033	.0239919	.0241812
32	.0243712	.0245619	.0247535	.0249457	.0251387	.0253325	.0255269	.0257222
33	0259182	.0261149	.0263124	.0265106	.0267095	.0269092	.0271097	.0273109
34	.0275128	.0277155	.0279189	.0281231	.0283279	.0285336	.0287399	.0289471
35	.0291550	.0293636	.0295729	.0297831	.0299939	.0302056	.0304179	.0306309
36	.0308448	.0310594	.0312747	.0314908	.0317075	.0319251	.0321434	.0323624
37	.0325822	.0328027	.0330239	.0332460	.0334687	.0336922	.0339165	.0341415
38	.0343672	.0345937	.0348209	.0350489	.0352775	.0355070	.0357372	.0359681
39	.0361998	.0364322	.0366654	.0368993	.0371339	.0373694	.0376055	.0378424
40	.0380800	.0383184	.0385575	.0387973	.0390379	.0392793	.0395214	.0397642
41	.0400078	.0402521	.0404972	.0407430	.0409895	.0412368	.0414849	.0417337
42	.0419832	.0422335	.0424845	.0427362	.0429887	.0432420	.0434959	.0437507
43	.0440062	.0442624	.0445194	.0447771	.0450355	.0452947	.0455547	.0458154
44	.0460768	.0463389	.0466019	.0468655	.0471299	.0473951	.0476609	.0479276
45	.0481950	.0484631	.0487320	.0490016	.0492719	.0495430	.0498149	.0508875
46	.0503608	.0506349	.0509097	.0511853	.0514615	.0517386	.0520164	.0522949
47	.0525742	.0528542	.0531349	.0534165	.0536988	.0539818	.0542655	.0545499
48	.0548352	.0551212	.0554079	.0556953	.0559835	.0562725	.0565622	.0568526
49	.0571438	.0574357	.0577284	.0580218	.0583159	.0586109	.0589065	.0592029
50	.0595000	.0597979	.0600965	.0603959	.0606959	.0609969	.0612984	.0616007
51	.0619038	.0622076	.0625122	.0628175	.0632235	.0634304	.0637379	.0640462
52	.0643552	.0646649	.0649753	.0652867	.0655987	.0659115	.0662250	.0665392
53	.0668542	.0671699	.0674864	.0678036	.0681215	.0684402	.0687597	.0690799
54	.0694008	.0697225	.0700449	.0703681	.0705293	.0710166	.0713419	.0716681
55	.0719950	.0724226	.0726510	.0729801	.0733099	.0736406	.0739719	.0743039
56	.0746368	.0749704	.0753047	.0756398	.0759755	.0763120	.0766494	.0769874
57	.0773262	.0776657	.0780060	.0783476	.0786887	.0790312	.0793745	.0797185
58	.0800632	.0804087	.0807549	.0811019	.0814495	.0817980	.0821472	.0824971
59	.0828478	.0831992	.0835514	.0839043	.0842579	.0846123	.0849675	.0853234
60	.0856800	.0860374	.0863955	.0867543	.0871139	.0874743	.0878354	.0881973

The result is only approximately correct on account of the area of the piston rod not being deducted from the area of the piston on the crank end. It is sufficiently accurate, however, for practical purposes.

Brake Horsepower. All of the i.h.p. is not available for useful work as the internal friction of the engine absorbs some of the energy, so the net horsepower is the i.h.p. less the h.p. consumed by the engine in overcoming internal resistances. This net horsepower is usually spoken of as the brake horsepower (b.h.p.) and it is obtained by the use of some form of brake.

Mechanical Efficiency. The mechanical efficiency of an engine is the ratio between the b.h.p. and the i.h.p. Expressed in per cent, it would be $\frac{\text{b.h.p.} \times 100}{\text{i.h.p.}}$. It is sometimes given as the engine friction in per cent, that is, the mechanical efficiency is expressed as ten or fifteen per cent engine friction, which is evidently 100 minus the mechanical efficiency. Under ordinary conditions, the engine friction varies from about 6 to 10 per cent of the i.h.p. depending on the size and the construction of the engine.

Piston Displacement. The piston displacement is the space in the cylinder swept through by the piston in its travel, expressed in cubic feet. The piston displacement for the c.e. will be less than for the h.e. by an amount equal to the area of the piston rod in square feet multiplied by the stroke in feet. In the 10- by 16-inch engine with $1\frac{3}{4}$-inch piston rod, the piston displacement for head end is

$$\underset{\text{(Head End)}}{\text{Piston displacement}} = \frac{.7854 \times 10^2 \times 16}{1728}$$
$$= .72722 \text{ cubic feet}$$

For the crank end, it would be .72722 less the cubic feet in the piston rod or

$$\underset{\text{(Crank End)}}{\text{Piston displacement}} = .72722 - \left(\frac{.7854 \times 1.75^2 \times 16}{1728}\right)$$
$$= .72722 - .02221$$
$$= .70501 \text{ cubic feet}$$

PROPERTIES OF STEAM

Saturated Vapor. The process of converting a liquid into a vapor is known as vaporization; the product thus formed is readily condensed and, therefore, does not follow the laws of perfect gases. A dry saturated vapor is one that has just enough heat in it to keep

it in the form of a vapor; if we add more heat, it becomes superheated. A superheated vapor may lose a part of its heat without condensation; a saturated vapor can not. When a saturated vapor loses a part of its heat, some of it will condense and we say that the vapor is wet.

Steam is simply the vapor from water and we shall confine our discussion to this alone. Suppose we have a vertical cylinder, as shown in Fig. 38, fitted with a light piston free to move up and down, yet so constructed that it may be loaded at will. Suppose that there is one pound of water at a temperature of 32°F. in the bottom of cylinder *A*, and that the piston rests upon its surface. Now, if heat is applied by means of a gas flame or fire, we shall notice the following effects:

(1) The temperature of the water will gradually rise until it reaches the temperature at which steam is formed. This temperature will depend upon the pressure, or the load on the piston. If the piston is very light, we may neglect its weight and consider that there is simply the atmospheric pressure of 14.7 pounds per square inch acting on the water surface, at which pressure, steam forms at 212° F.

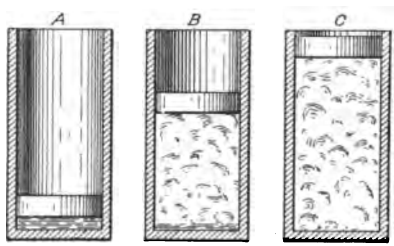


Fig. 38. Engine Cylinder Containing Water and Steam

(2) Therefore, as soon as 212 degrees is reached, steam will form and the piston will steadily rise (*B*, Fig. 38), but no matter how hot the fire may be, the temperature of both water and steam will remain at 212 degrees until all the water is evaporated (*C*, Fig. 38).

We had one pound of water at 32 degrees and at 14.7 pounds absolute pressure, and found that steam formed at a temperature of 212 degrees and remained at that temperature. There were added 180.3 B.T.U., the heat of the liquid, to bring the water from 32 degrees to the boiling point. To convert water at 212 degrees into steam at 212 degrees, there were added 969.7 B.T.U. more. This quantity, known as the latent heat, or heat of vaporization, makes the total heat 1150.0 B.T.U. If we should measure the volume carefully after all the water was evaporated, we should find

that there were exactly 26.78 cubic feet of dry saturated steam. At the start we had one pound of water and, therefore, we must have one pound of steam, for none could escape; hence one cubic foot will weigh $\frac{1}{26.78}$, or 0.03734 pound, which is known as the density of steam at 14.7 pounds absolute pressure and 212° F.

Effect of Pressure on Boiling Point. Suppose now we place a weight of 85.3 pounds per square inch on the piston. The pressure is 85.3 plus 14.7 or 100 pounds per square inch absolute. We shall now find that no steam will form until a temperature of 327.86 degrees is reached, and also we must add 887.6 B.T.U. Under this greater pressure the steam occupies a volume of only 4.432 cubic feet, or one cubic foot of it weighs $\frac{1}{4.432}$, or 0.2256 pound.

From the foregoing, it is obvious that there is a definite relation between the pressure and the temperature, that is, the temperature at which water will boil depends upon the pressure, and *vice versa.*

Of course, it would be impossible to determine all these different quantities by actual experiment, and at all pressures varying from vacuum to the high pressures used in water-tube boilers; they can be computed.

Steam Tables. We have already seen that any change in the temperature of saturated steam produces a change of pressure, and that every change of pressure corresponds to a certain change in temperature. There are several properties of saturated steam that depend upon the temperature and pressure; and the values of all these different properties when arranged for all temperatures and pressures are called Steam Tables. The following are the principal items found in the tables:

(1) *Absolute pressure* in pounds per square inch; it is equal to the gauge pressure plus the atmosphere pressure of 14.7 pounds or the pressure of atmosphere as obtained from the barometer.

(2) *Temperature of steam,* or of boiling water, at the corresponding pressure.

(3) *Heat of liquid;* or the number of B.T.U. necessary to raise one pound of water from 32°F. to the boiling point corresponding to the given pressure.

(4) *Heat of vaporization,* or latent heat; that is, the number of B.T.U. necessary to change one pound of water, at the boiling point, into dry satur-

ated steam at the same temperature and pressure. It was noted that in heating water from 32°F. to the boiling point under atmosphere pressure, the temperature rose from 32 degrees to 212 degrees, but as soon as 212 degrees was reached, the temperature remained constant until all the water was converted into steam at that pressure. While the process of changing the water at 212 degrees into steam at 212 degrees was going on, heat was being added but no change occurred in temperature. This phenomenon always occurs when the application of heat results in the change of state of a substance, either from a solid to a liquid, or from a liquid to a gaseous state. This apparent loss of heat is not real, but recurs whenever the transformation is reversed, that is when the steam is condensed.

(5) *Total heat of vaporization* is the number of B.T.U. necessary to change one pound of water from 32°F. into steam at the given temperature or pressure, and represents the sum of the heat units of the liquid plus the heat units of vaporization or the latent heat.

(6) *Density of steam,* which is the weight of one cubic foot of steam at the given temperature or pressure.

(7) *Specific volume,* which is the volume occupied by one pound of steam. The specific volume is the reciprocal of the density, that is, it is equal to $\frac{1}{\text{density}}$.

Steam tables from which the above items may be obtained are very useful to the engineer. Any one wishing to make a more detailed study of the steam tables and the properties of steam should procure "Steam and Entropy Tables" by Peabody, published by John Wiley & Sons.

Kinds of Steam. If the process of adding heat to water and then to steam be continued, three kinds of steam are obtained depending on the conditions, namely, saturated or dry steam, wet steam, and superheated steam.

Saturated or Dry Steam. If just sufficient heat be added to the vessel A, Fig. 38, until the pound of water is completely converted into steam as shown in C, the result is a pound of saturated steam. The number of heat units that have been added equals the heat of the liquid q plus the latent heat r. Therefore Q, the number of B.T.U. added, in equation form becomes

$$Q=q+r$$

It is evident, therefore, that saturated steam contains all the heat of the liquid plus the heat of vaporization.

Wet Steam. If instead of adding enough heat to the vessel C to completely evaporate the water, the operation be discontinued while there is yet some water remaining unevaporated, as is shown

TABLE III
Properties of Saturated Steam

Total pressure in lbs. per sq. in. above vacuum p	Temperature in degrees Fahrenheit t	Heat in liquid from 32° in heat units q	Heat of vaporization or latent heat in heat units r	Total heat in heat units from water at 32° H	Density or weight of one cubic foot in lbs. $\frac{1}{s}$	Volume of 1 pound in cubic feet s	Total pressure in lbs. per sq. in. above vacuum p
1	101.84	69.8	1034.7	1104.5	0.00300	333.1	1
2	126.15	94.2	1021.9	1116.1	0.00578	173.1	2
3	141.52	109.6	1012.2	1121.8	0.00845	118.4	3
4	153.00	121.0	1005.5	1126.5	0.01106	90.4	4
5	162.26	130.3	1000.0	1130.3	0.01364	73.3	5
6	170.07	138.1	995.5	1133.6	0.01616	61.9	6
7	176.84	144.9	991.4	1136.3	0.01866	53.6	7
8	182.86	150.9	987.8	1138.7	0.02116	47.26	8
9	188.27	156.4	984.5	1140.9	0.02362	42.36	9
10	193.21	161.3	981.4	1142.7	0.02606	38.37	10
14.7	212.00	180.3	969.7	1150.0	0.03734	26.78	14.7
15	213.03	181.3	969.1	1150.4	0.03805	26.28	15
20	227.95	196.4	959.4	1155.8	0.04978	20.09	20
25	240.07	208.7	951.4	1160.1	0.06140	16.29	25
30	250.34	219.1	944.4	1163.5	0.0728	13.74	30
35	259.29	228.2	938.2	1166.4	0.0842	11.88	35
40	267.26	236.4	932.6	1169.0	0.0953	10.49	40
45	274.46	243.7	927.5	1171.2	0.1065	9.387	45
50	281.03	250.4	922.8	1173.2	0.1176	8.507	50
55	287.09	256.6	918.4	1175.0	0.1286	7.778	55
60	292.74	262.4	914.3	1176.7	0.1395	7.166	60
65	298.00	267.8	910.4	1178.2	0.1504	6.647	65
70	302.96	272.9	906.6	1179.5	0.1613	6.199	70
75	307.64	277.7	903.1	1180.8	0.1722	5.807	75
80	312.08	282.2	899.8	1182.0	0.1829	5.466	80
85	316.30	286.5	896.6	1183.1	0.1938	5.161	85
90	320.32	290.7	893.5	1184.2	0.2047	4.886	90
95	324.16	294.6	890.5	1185.1	0.2153	4.644	95
100	327.86	298.5	887.6	1186.1	0.2256	4.432	100
105	331.42	302.1	884.8	1186.9	0.2362	4.233	105
110	334.83	305.6	882.1	1187.7	0.2471	4.047	110
115	338.14	309.0	879.5	1188.5	0.2580	3.876	115
120	341.31	312.3	876.9	1189.2	0.2686	3.723	120
125	344.39	315.5	874.5	1190.0	0.2793	3.581	125
130	347.38	318.6	872.1	1190.7	0.2898	3.451	130
140	353.09	324.4	867.4	1191.8	0.3106	3.220	140
150	358.50	330.0	863.0	1193.0	0.3318	3.014	150
160	363.62	335.3	858.8	1194.1	0.3528	2.834	160
170	368.50	340.4	854.8	1195.2	0.3741	2.673	170
180	373.16	345.2	850.9	1196.1	0.3951	2.531	180
190	377.61	349.8	847.1	1196.9	0.4158	2.405	190
200	381.89	354.3	843.5	1197.8	0.4371	2.288	200
210	386.02	358.6	840.0	1198.6	0.4579	2.184	210
220	389.98	362.7	836.6	1199.3	0.4789	2.088	220
230	393.80	366.6	833.3	1199.9	0.4997	2.001	230
240	397.50	370.5	830.1	1200.6	0.521	1.921	240
250	401.10	374.2	826.9	1201.1	0.542	1.845	250
260	404.55	377.8	823.9	1201.7	0.563	1.775	260
270	407.90	381.3	820.9	1202.2	0.584	1.711	270
280	411.19	384.8	818.0	1202.8	0.605	1.652	280
290	414.35	388.1	815.2	1203.3	0.627	1.595	290
300	417.45	391.3	812.4	1203.7	0.649	1.542	300
310	420.45	394.4	809.7	1204.1	0.670	1.492	310
320	423.40	397.5	807.1	1204.6	0.692	1.446	320

TABLE IV
Properties of Saturated Steam

Temperature in degrees Fahrenheit t	Total pressure above vacuum p	Heat in liquid in heat units from 32° q	Heat of vaporization or latent heat in heat units r	Total heat in heat units from water at 32° H	Density or weight of one cubic foot in lbs. $\frac{1}{s}$	Volume of 1 pound in cubic feet s	Temperature in degrees Fahrenheit t
32	0.0886	0.0	1071.7	1071.7	0.000302	3308.0	32
60	0.2561	28.1	1057.0	1085.1	0.000828	1207.0	60
90	0.6960	58.1	1041.2	1099.3	0.002131	469.2	90
120	1.689	88.0	1024.4	1112.4	0.004926	203.	120
140	2.885	108.0	1013.1	1121.1	0.00814	122.8	140
150	3.715	118.0	1007.2	1125.2	0.01032	96.9	150
160	4.738	128.0	1001.4	1129.4	0.01296	77.2	160
170	5.990	138.0	995.5	1133.5	0.01613	62.0	170
180	7.510	148.0	989.5	1137.5	0.01993	50.2	180
190	9.339	158.1	983.4	1141.5	0.02444	40.92	190
200	11.528	168.2	977.2	1145.4	0.02974	33.62	200
212	14.698	180.3	969.7	1150.0	0.03734	26.78	212
220	17.188	188.4	964.6	1153.0	0.04321	23.14	220
225	18.914	193.4	961.4	1154.8	0.04726	21.16	225
230	20.780	198.5	958.1	1156.6	0.05160	19.37	230
235	22.790	203.6	954.8	1158.4	0.05630	17.77	235
240	24.970	208.6	951.4	1160.0	0.06130	16.31	240
245	27.310	213.7	948.1	1161.8	0.06660	15.01	245
250	29.82	218.8	944.7	1163.5	0.0724	13.82	250
255	32.53	223.8	941.2	1165.0	0.0785	12.73	255
260	35.42	229.0	937.8	1166.8	0.0851	11.75	260
265	38.53	234.0	934.3	1168.3	0.0920	10.87	265
270	41.84	239.1	930.7	1169.8	0.0995	10.05	270
275	45.39	244.2	927.2	1171.4	0.1074	9.309	275
280	49.19	249.4	923.6	1173.0	0.1158	8.639	280
285	53.22	254.5	920.0	1174.5	0.1247	8.021	285
290	57.53	259.6	916.3	1175.9	0.1341	7.454	290
295	62.11	264.7	912.6	1177.3	0.1441	6.937	295
300	66.98	269.8	908.9	1178.7	0.1547	6.462	300
305	72.15	274.9	905.1	1180.0	0.1660	6.024	305
310	77.63	280.1	901.3	1181.4	0.1779	5.622	310
315	83.44	285.2	897.6	1182.8	0.1903	5.254	315
320	89.59	290.4	893.7	1184.1	0.2038	4.907	320
325	96.12	295.5	889.8	1185.3	0.2177	4.594	325
330	102.98	300.6	885.9	1186.5	0.2319	4.312	330
335	110.25	305.8	882.0	1187.8	0.2476	4.038	335
340	117.91	310.9	878.0	1188.9	0.2642	3.784	340
345	126.00	316.1	874.0	1190.1	0.2813	3.554	345
350	134.52	321.3	870.0	1191.3	0.2992	3.342	350
355	143.46	326.4	865.9	1192.3	0.3178	3.147	355
360	152.89	331.6	861.8	1193.4	0.3378	2.960	360
365	162.77	336.8	857.7	1194.5	0.3588	2.787	365
370	173.17	341.9	853.5	1195.4	0.3808	2.626	370
375	184.08	347.1	849.3	1196.4	0.4035	2.478	375
380	195.52	352.3	845.1	1197.4	0.4275	2.339	380
385	207.49	357.5	840.8	1198.3	0.4527	2.209	385
390	220.05	362.7	836.6	1199.3	0.4789	2.088	390
395	233.20	367.9	832.2	1200.1	0.5060	1.975	395
400	246.9	373.1	827.9	1201.0	0.535	1.868	400
405	261.3	378.3	823.5	1201.8	0.566	1.766	405
410	276.3	383.5	819.1	1202.6	0.598	1.673	410
415	292.0	388.7	814.6	1203.3	0.631	1.584	415
420	308.5	394.0	810.1	1204.1	0.667	1.499	420
425	325.6	399.2	805.6	1204.8	0.704	1.421	425

in the bottom of B, the result would be steam with some water in suspension. Steam in this state is known as wet steam, that is, it contains some moisture. Expressed in equation form, the number of B.T.U. added would be

$$Q = q + xr$$

in which x = the weight of the part vaporized. By means of a steam calorimeter, the amount of water held in suspension by the steam may be determined. In practice, it amounts to about one to three per cent, depending upon the mechanical construction of the plant and will average about two per cent. The quality of steam, considering saturated steam as unity or one, would then be $100-2=98$ per cent, dry. So the quantity x is the 98 per cent or the per cent of the total amount of water that has been vaporized.

Superheated Steam. If after saturated or dry steam is obtained, additional heat be added by some means until the temperature of the dry steam is above that corresponding to the pressure, it is said to be superheated. In obtaining superheated steam, more B.T.U. have been added than when dry steam was obtained, so another expression is used to represent the total heat B.T.U. added, viz,

$$Q = q + r + .48\ (t_s - t)$$

in which t_s equals the temperature of the superheated steam in degrees F. and is obtained by the use of thermometers; t equals the temperature corresponding to the absolute boiler pressure; and .48 equals the specific heat of superheated steam at constant pressure. This factor varies slightly for different pressures and temperatures, but for general use the value given is sufficiently accurate. It may be obtained at any pressure and temperature by experiment.

It is obvious from the foregoing that the number of B.T.U. contained in one or more pounds of steam, be it wet, dry, or superheated, can be obtained by the use of the above formulas and the steam tables.

In order to become familiar with the above formulas and the use of the steam tables, a few simple problems will be worked out. It must be borne in mind in the use of these tables that whenever a pressure is given, the other properties, such as t, q, r, etc., are found in Table III; and that if the temperature be given, Table IV must

be used. It is also to be remembered that the tables are based on absolute pressures, so if the gauge pressure be given instead of the absolute pressure, the gauge pressure reading must be converted into absolute pressure by adding the atmospheric pressure. For example, if 160 pounds gauge pressure, say at sea level, is given instead of 160 pounds absolute, then before looking for t, r, etc., corresponding to that pressure, 14.7 pounds should be added to the 160 pounds, making the absolute pressure 174.7 pounds. If the barometric pressure is 29.4 inches of mercury when the gauge pressure is 160 pounds, then the absolute pressure will be $160+(29.4\times.49)=160+14.4$, or 174.4 pounds per square inch. This must always be done before making use of the steam tables.

ILLUSTRATIVE PROBLEMS

EXAMPLE 1. How many heat units in one pound of water at 160°F.?

SOLUTION. Looking down the first column of Table IV until 160 degrees is found, then following across horizontally, we find in the third column under Heat of the Liquid 128.0 B.T.U., which is the number of heat units contained in one pound of water at 160°F.

EXAMPLE 2. What temperature corresponds to 160 pounds absolute?

SOLUTION. Since the tables are based on absolute pressures and the pressure of 160 pounds is given as absolute, we turn to Table III and follow down the first column until 160 pounds is reached, then horizontally across to the second column where we find the temperature corresponding to 160 pounds absolute to be 363.62°F.

EXAMPLE 3. What is the heat of vaporization r at 160°F.?

SOLUTION. Since it is temperature that is given, it is necessary to find 160 degrees in the first column of Table IV and following across horizontally to the fourth column, we find that the heat of vaporization r is 1001.4 B.T.U.

EXAMPLE 4. What is the value of r for 160 pounds absolute pressure?

SOLUTION. Since the pressure is given, it is necessary to look in the first column of Table III for 160 pounds, and following across to the fourth column we find r to be 858.8 B.T.U.

EXAMPLE 5. Steam is made in a boiler at 140 pounds per square inch absolute from feed water at a temperature of 70°F., 99 per cent of each pound being evaporated. How many heat units are spent in raising the temperature of one pound of the water to the boiling point? What are the total number of B.T.U. required to make one pound of steam?

SOLUTION. Looking for q in Table IV, corresponding to the temperature of 70°F., we find that it is necessary to interpolate between 90 degrees and 60 degrees. For 90 degrees, q is 58.1; for 60 degrees it is 28.1. The difference is $58.1-28.1$, or 30.0 for a difference of 30 degrees. For one degree, the value would be $\frac{30.0}{30}$, or 1.0. Since there is a difference of 10 degrees between 60 degrees and the temperature of the feed water, we must add to q for 60

degrees 1.0×10, or 10.0, making $28.1+10.0$, or 38.1. This is the required q for 70 degrees, or the number of B.T.U. in one pound of the feed water as it enters the boiler. Next, obtain the number of B.T.U. in one pound of the water in the boiler after being raised to 140 pounds absolute pressure. In Table III the q corresponding to 140 pounds absolute pressure is found to be 324.4 B.T.U. Since the feed water contained 38.1 B.T.U., the number of B.T.U. that has been added in raising one pound of the water from 70 degrees to 140 pounds pressure is $324.4-38.1$, or 286.3 B.T.U., the required result.

Since the steam formed contains some moisture, its quality being 99 per cent, it follows that Q, the number of B.T.U. required to vaporize one pound of the water under the given conditions, would be $q+xr$. In the first part of the problem q was found to equal 286.3 B.T.U. The value of r for 140 pounds absolute, as obtained from Table III, is 867.4.

$$\begin{aligned} Q &= 286.3+(.99\times867.4) \\ &= 286.3+858.73 \\ &= 1145.03 \text{ B.T.U.} \end{aligned}$$

This result represents the total heat units necessary to make one pound of steam under the given conditions.

Feed Water Temperature. The above problem brings out a point that has not been noted, viz, the method to follow when the temperature of the feed water is other than 32 degrees, q in the equation $Q=q+x\,r$ being the heat of the liquid corresponding to the pressure, taking 32 degrees as standard. When the feed water at the outset is of a higher temperature than 32 degrees, it is obvious that on account of this higher temperature the number of heat units required to raise it to the boiling point will be less. It follows, therefore, that the above expression should be modified in order to apply to feed water of any temperature. That is, if t is equal to the temperature of the feed water from which the steam is to be made, and q_1 equals the corresponding heat of the liquid, the expression for Q may be modified so as to read

$$Q=q+x\,r-q_1$$

or, taking $q+x\,r=H$, the total amount of heat units added is

$$Q=H-q_1$$

Likewise, the formula for superheated steam may be changed to the form

$$\begin{aligned} Q &= q+r+.48\,(t_s-t)-q_1 \\ &= H+.48\,(t_s-t)-q_1 \end{aligned}$$

Example. How many B.T.U. must be added to one pound of water at 177°F. to transform it into steam at 145.5 pounds gauge pressure and a temperature of 480° F.?

SOLUTION. From inspection, it is evident that the result will be superheated steam, since 480° F. is higher than the temperature corresponding to the 145.5 pounds gauge pressure. This being true, the above formula for superheated steam must be used. Assuming the atmosphere pressure to be 14.5 pounds per square inch, the absolute pressure will be 160 pounds. The value of q corresponding to this pressure is 335.3, r is 858.8, t_s is 480 degrees, t is 363.62, and q_1 is 145.0. Substituting these values in the above formula, we have

$$Q = 335.3 + 858.8 + .48\,(480 - 363.62) - 145.0$$
$$= 1194.1 + 55.86 - 145.0 = 1104.96 \text{ B.T.U}$$

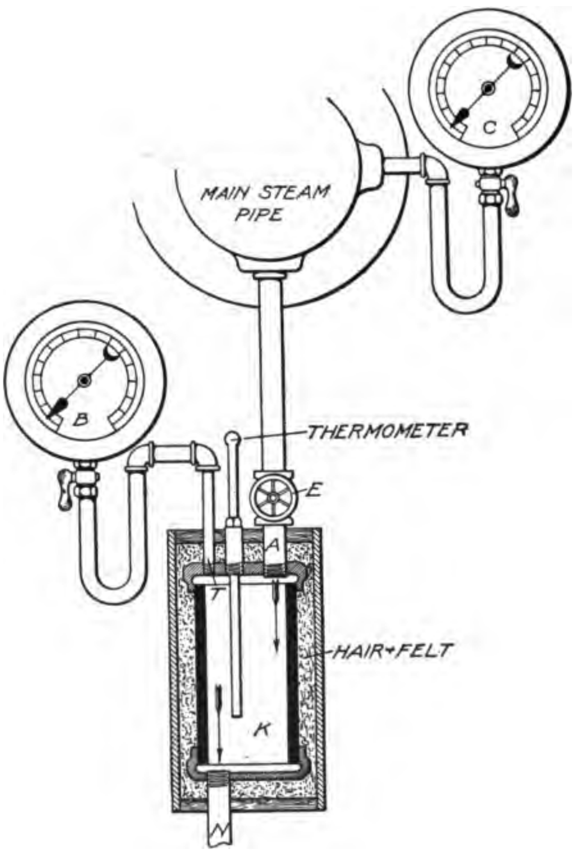


Fig. 39. Throttling Calorimeter Connected for a Test

Calorimetric Measurements. In the discussion of the properties of steam and the use of the steam tables, the term *quality of steam* was referred to and was used in every instance where saturated

steam was dealt with. Since it is necessary that the quality of steam be known in all calculations dealing with the amount of heat in a pound of steam, some means must be employed for determining the quality.

Throttling Calorimeter. An apparatus known as a throttling calorimeter was devised in the early eighties by Professor C. H. Peabody for making this determination. It has been widely used since that time and is considered a simple and efficient means for obtaining the required results.

The operation of the throttling calorimeter is based upon the principle that saturated steam will become superheated if the pressure is reduced by throttling without loss of heat. The calorimeter, Fig. 39, performs the above function within certain limits, as will be evident from a description of its action. It consists of a closed metallic cylinder *K*, having a steam inlet *A* and an outlet *N;* a thermometer well, made of suitable material, is provided at *T*. Two gauges *B* and *C* are used, *B* being attached to the calorimeter and *C* to the steam supply line by means of siphons. A valve *E* is placed between the main steam pipe and the calorimeter so as to regulate the amount of flow of steam into the calorimeter. The nipple *A*, connecting the inlet valve *E* with the chamber *K*, is made of special metal, threaded, and having a well-formed orifice, as shown in Fig. 40.

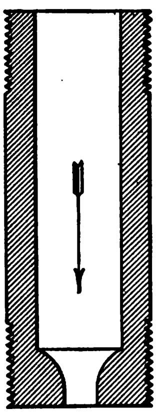


Fig. 40. Form of Connecting Nipple

The connection between the steam pipe and the calorimeter should be as short as possible. The cylinder *K* and the connections should be thoroughly covered with asbestos hair felt, or other nonconducting material, in order to reduce the amount of heat radiation. The outlet pipe *N* should be at least 1 inch in diameter for its entire length and it may be larger.

To use the calorimeter, fill the thermometer well *T* with oil or mercury and then insert the thermometer. Attach the gauges and permit steam to enter the cylinder *K* until, say, about 5 pounds pressure is registered on the gauge *B*. This pressure should be kept constant throughout the test by means of the valve *E*. The siphons should be full of water and steam should be permitted to flow through

the apparatus for about ten minutes before taking observations. The observations to be taken are the pressure P of the steam in the main steam pipe, the pressure P_1 of the steam in the calorimeter, the temperature t in the calorimeter, and the barometric pressure P_a. Having this data at hand, the amount of moisture in the steam may be determined by combining the two fundamental equations $Q=q+xr$, corresponding to P, the main steam pipe pressure, and $Q=q_1+r_1+C_p\ (t_s-t_1)$, corresponding to the pressure P_1 in the calorimeter. The absolute pressure in the main steam pipe will be $P+P_a$, and in the calorimeter P_1+P_a. Equating these two expressions, we get

$$q+xr=q_1+r_1+C_p\ (t_s-t)$$

Transposing and dividing through by r, we get

$$x=\frac{q_1+r_1+C_p\ (t_s-t)-q}{r}$$

x being 1 minus the per cent of moisture in the steam, or the quality.

The equation and the method of obtaining the quality of steam will be readily understood by the following example.

Example. The pressure P in the main steam pipe equals 69.8 pounds; the pressure P_1 in the calorimeter equals 12 pounds; the pressure P_a of the atmosphere equals 14.8 pounds, the temperature t_s in the calorimeter equals 268.2° F. Determine the quality of the steam.

Solution. The absolute pressure in the steam pipe $P+Pa=69.8+14.8$, or 84.6 pounds. The absolute pressure in the calorimeter $P_1+Pa=12+14.8$, or 26.8 pounds. The temperature of saturated steam t_1 at 26.8 pounds $=243.8$ pounds. It is to be noted that t_1 is the temperature corresponding to the absolute pressure in the calorimeter. The total heat $q_1+r_1=1161.3$ B.T.U.; $q=$ the heat of the liquid corresponding to $P+Pa=286.2$, and r for the same pressure is 896.9.

$$\therefore \quad x=\frac{1161.3+0.48\ (268.2-243.8)-286.2}{896.9}$$
$$=.989$$

A throttling calorimeter may be made of pipe fittings, making a simple and convenient apparatus which, if properly constructed and operated, will give good results. Such an apparatus is illustrated in Fig. 41, which also shows the proper method of connection to a steam pipe. Steam is taken from a ½-inch pipe provided with a valve and passes through two ¾-inch tees situated on opposite sides

of a ¾-inch flange union, substantially as shown in the accompanying sketch. A thermometer cup, or well, is screwed into each of these tees, and a piece of sheet iron perforated with a ⅛-inch hole in the center is inserted between the flanges and made tight with rubber or asbestos gaskets, which also act as nonconductors of heat. For convenience a union is placed near the valve as shown, and the exhaust steam may be led away by a short 1¼-inch pipe, shown by dotted lines. The thermometer wells are filled with mercury or heavy cylinder oil, and the whole instrument from the steam main to the 1¼-inch pipe is well covered with hair felt.

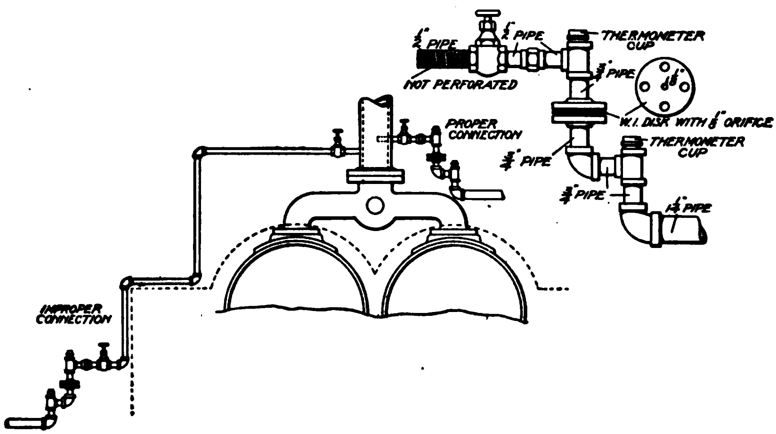


Fig. 41. Throttling Calorimeter Made of Pipe Fittings

Great care must be taken that the ⅛-inch orifice does not become choked with dirt, and that no leaks occur, especially at the sheet iron disk, also that the exhaust pipe does not produce any back pressure below the flange. Place a thermometer in each cup, and opening the ½-inch valve wide, let steam flow through the instrument for ten or fifteen minutes; then take frequent readings on the two thermometers and the boiler gauge, say at intervals of one minute.

Separating Calorimeter. Another type of calorimeter sometimes used in cases where the steam contains from 5 to 10 per cent of moisture, is the separating calorimeter. It works upon the principle that the moisture contained in the steam is liberated by mechanical means. In its usual form, the calorimeter consists of a cylindrical vessel so constructed that the moisture is separated from the

steam and returned, the dry steam passing to a condenser where it is collected and afterward weighed. The separating vessel is provided with a glass gauge and graduated scale which indicates the weight of the moisture taken out of the steam. Having obtained the weight W of the separated water and the weight W_1 of the dry steam, then the percentage of moisture to the total amount of the liquid would be

$$y=\frac{W}{W+W_1}$$

Therefore, the percentage of dry steam would be

$$x=1-y$$
$$=1-\frac{W}{W+W_1}$$

EXAMPLE. Required the weight of steam in the cylinder of a 16×36-inch engine when the piston has moved on its stroke 27.4 per cent of the distance from the h.e. The steam in the cylinder at this instant is under a pressure of 114 pounds gauge, as determined from the indicator card. The piston displacement for the h.e. is 4.188 cubic feet and the clearance on the h.e. is 5.2 per cent. The atmosphere pressure is 14.5 pounds per square inch.

SOLUTION. The first thing required is the volume of steam back of the piston, when the engine has made 27.4 per cent of the stroke. To 27.4 per cent add the clearance 5.2 per cent, making a total of 32.6 per cent, or $\frac{32.6}{100}$ of the whole volume of the cylinder containing steam at the instant under consideration. Since the total piston displacement for the h.e. is 4.188 cubic feet, then the volume of steam to be considered would be $4.188\times\frac{32.6}{100}$, or 1.36 cubic feet. The absolute pressure of the steam in the cylinder would be 114+14.5 pounds, or 128.5 pounds per square inch. Looking in Table III for the weight of 1 cubic foot of steam at 128.5 pounds absolute pressure, we find it to be by interpolation .2867 pounds. Since there are 1.36 cubic feet in the cylinder at 27.4 per cent of the stroke, the weight of steam in the cylinder at the instant in question would be 1.36×.2867, or .3899 pounds.

Volume and Weight of Steam. In considering any problem dealing with the weight of steam in the cylinder or the piston displacement, the per cent of clearance must always be taken into account, as in the problem above.

In studying and analyzing the performance of an engine, it is often desirable to obtain the volume and the weight of steam in the cylinder from the indicator card at the several events, and also to

know the quality of the steam at these points. From the study of the indicator cards and the steam tables, we are now prepared to obtain these several values.

Volume of steam at c.o. in cubic feet is equal to the piston displacement in cubic feet multiplied by the per cent of c.o. plus the per cent of clearance. For example, in the problem given above, the h.e. displacement was 4.188 cubic feet and the h.e. clearance was 5.2 per cent. It is desired to obtain the volume of steam at c.o. which takes place at 34.8 per cent. The sum of the clearance and c.o. per cents is 40. Therefore, the volume of steam at c.o. is 4.188×40 per cent, or 1.675 cubic feet.

Volume of steam at release in cubic feet is the product of the piston displacement and the sum of the per cents of release and clearance.

Volume of steam at compression is found by multiplying the piston displacement by the per cent of compression plus the per cent of clearance.

Weight of steam at c.o. is the product of the weight of 1 cubic foot of steam at the absolute pressure at c.o. and the volume of steam in cubic feet at c.o., both as obtained from the indicator card.

Weight of steam at release is the product of the weight of 1 cubic foot of steam at the absolute pressure at release and the volume of steam in cubic feet at release.

Weight of steam at compression is found in the same manner as that at release.

Re-evaporation or condensation per revolution in pounds is the weight of steam at release minus the weight of steam at c.o. If the answer is positive, it indicates that there is a re-evaporation, and if negative, a condensation.

Re-evaporation or condensation per i.h.p. per hour in pounds is the item just given multiplied by the revolutions per hour and divided by the i.h.p.

Weight of steam per revolution, as determined by weighing, is the total weight of steam used by the engine divided by the total number of revolutions.

Weight of mixture in the cylinder per revolution in pounds is the weight of steam per revolution as determined by weighing plus the weight of steam at compression.

Per cent of mixture accounted for as steam at c.o. is one hundred times the weight of steam at c.o. per revolution, divided by the weight of the mixture in the cylinder per revolution.

Per cent of mixture accounted for as steam at release is one hundred times the weight of steam at release per revolution divided by the weight of the mixture in the cylinder per revolution.

Thermal Efficiency. Having obtained a working knowledge of the properties of steam from the preceding discussion and the problems dealing with the B.T.U. values of steam, we are now ready to consider the thermal efficiency of an engine, but before this can be calculated, several things must be known.

(1) The amount of work done in a unit of time.
(2) The weight of steam used by the engine in the same length of time.
(3) The number of B.T.U. in each pound of steam used.

These quantities must be accurately determined while the engine is in operation.

EXAMPLE. To illustrate what is meant by thermal efficiency, assume an engine which in developing 242 h.p. uses 13,000 pounds of steam in two hours; steam pressure 186.3 pounds gauge; quality of steam 99 per cent; temperature of feed water 60°F.; and atmosphere pressure 14.7 pounds. Find the thermal efficiency in per cent.

SOLUTION. The number of foot pounds of work done in a minute is $242\times33{,}000=7{,}986{,}000$. The number of B.T.U. in one pound of steam at 186.3 pounds gauge, which is 200 pounds absolute, is $q+x\,r-q_1=354.3+(843.5\times.99)-28.1$, or 1161.27 B.T.U. Since the engine is using 13,000 pounds of steam in 2 hours, the amount of steam being used in one minute will be $\frac{13000}{2\times60}$, or 108.333. The corresponding number of B.T.U. supplied per minute will be 108.333×1161.27, or 125,804.25. Changing this to the equivalent foot pounds of energy by multiplying by 778, we get $125{,}804.25\times778$, or 97,875,706.5 as the total energy in foot pounds supplied the engine per minute. Therefore, the thermal efficiency is

$$E=\frac{\text{energy delivered per minute}\times100}{\text{energy supplied per minute}}$$

$$=\frac{7{,}986{,}000\times100}{97{,}923{,}444.58}=8.15\text{ per cent}$$

The thermal efficiency is expressed by some as the B.T.U. supplied per minute per i.h.p. instead of per cent. Applying this to the problem above, we get for the thermal efficiency

$$E=\frac{13000\times1161.27}{2\times60\times242}$$

$$=519.5\text{ B.T.U.}$$

Generally speaking, the efficiency of a steam engine is spoken of as being so many pounds of steam per i.h.p. per hour. In the problem under consideration, this would give an efficiency

$$E = \frac{13000}{2 \times 242}$$

$$= 26.8 \text{ pounds of steam per i.h.p. per hour}$$

The B.T.U. per i.h.p. per minute varies inversely as the thermal efficiency, so if one value is known, the other can be easily obtained by using the two constants—778 the mechanical equivalent of heat, and 33,000 the number of foot pounds per minute which constitutes a horsepower. If an engine has a thermal efficiency of 100 per cent, it would require $\frac{33000}{778}$, or 42.42 B.T.U. per h.p. per minute. An engine which used 520.1 B.T.U. per minute, as in the above example, has a thermal efficiency of $\frac{42.42}{519.5} \times 100$, or 8.17 per cent.

INTERPRETATION OF INDICATOR CARDS

Theoretical Diagram. As a basis of comparison between indicator diagrams taken from the same engine under different conditions, from different engines, and for design purposes, a theoretical diagram is constructed on the assumption that the expansion curve of a theoretically perfect engine would be that of a hyperbola. Experiments conducted at various times and on a large number of engines substantiate the assumption. The hyperbolic curve has the property that the product of the distances of any point on the curve from the line of zero volume is constant. This when expressed in equation form is

$$P_1 V_1 = C \text{ (constant)}$$

in which P_1 is pressure at c.o. and V_1 is volume at c.o. If P_2 is pressure at release and V_2 is volume at release, then

$$P_2 V_2 = C \text{ (constant)}$$

It follows then that $P_1 V_1 = P_2 V_2$. In this equation, P_1 and P_2 are absolute pressures and V_1 and V_2 include the clearance volume.

To Draw the Theoretical Card. To draw an ideal diagram (see Fig. 42), draw PX equal to the length of stroke and OP equal to the clearance. Draw OY and PA perpendicular to OX and draw YS parallel to OX and at a height corresponding to the boiler pressure.

The line of initial pressure AC is then drawn parallel to YS and is usually taken as from 90 to 95 per cent of the boiler pressure, if there is no special cause for loss. Then take AC as the portion of the stroke at which steam is admitted, so that $\frac{OX}{OR}$ equals the ratio of expansion. The expansion line is considered a hyperbolic curve with OY and OX as asymptotes. To draw the hyperbolic curve,

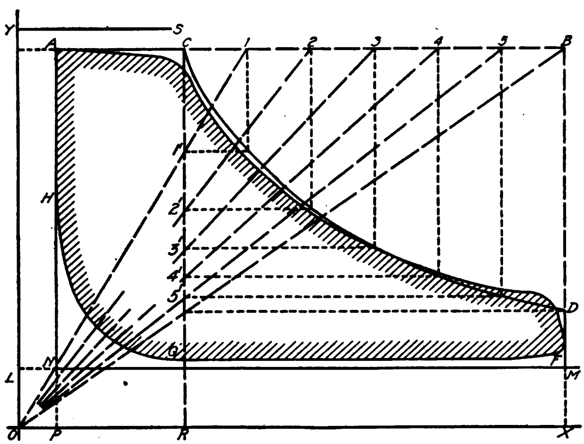


Fig. 42. Ideal Indicator Card

first draw the line ACB parallel to the atmosphere line and FDB and RC perpendicular to it. Then make points *1, 2, 3, 4,* etc., on CB and connect them with the point O. At the points *1′, 2′, 3′, 4′,* etc., where these lines intersect the line RC, draw parallels to CB until they meet perpendiculars from points *1, 2, 3, 4,* etc. The points of intersection of these lines are points on the hyperbolic curve CD, as shown in Fig. 42. Any number of points may be used, but there must be enough to determine the curve. A theoretical compression curve may be drawn in the same manner as an expansion curve, letting the perpendicular to the atmosphere line be drawn from the point of compression instead of from the point of cut-off.

The area $ACDMNH$ is the theoretical card, with a given

boiler pressure and an assumed drop and ratio of expansion. The actual card for the same data would probably appear more nearly like the shaded area which lies mostly within the outline of the theoretical card. In designing engines, it is well to know the ratio of the actual to the ideal card for all types of engines. This ratio varies between .5 and .9 according to the speed, type of engine, and kind of valves.

It will be observed that the actual expansion curve does not coincide with the theoretical curve in Fig. 42. It is a well-known fact that, in the cylinder of a steam engine, the temperature of the steam changes during the stroke. Usually, the piston and valves leak steam more or less; initial condensation takes place at the beginning of the stroke and re-evaporation at the end of the stroke. These factors cause a variation from the true theoretical curve. The object, therefore, in constructing the theoretical diagram is to ascertain where and to what extent these variations occur and to study the causes of the irregularities to the end that the necessary adjustments may be made to eliminate the errors in so far as possible.

In the construction of the theoretical indicator diagram, it is assumed that no loss of heat occurs in the cylinder. It is a well-known fact that as the steam enters the cylinder, some is condensed on account of the comparatively cool cylinder walls. Toward the end of the stroke, the cylinder walls give off heat with the result that either all or a part of the condensed steam is re-evaporated. Hence, the expansion curve of the theoretical diagram would naturally fall below that of the actual curve near the end of the stroke. Speaking in general, a close approximation of the two curves is an indication of good valve adjustment and economical steam distribution. It is, therefore, advantageous to draw the theoretical diagram in order to have something upon which to base an opinion as to the condition of the engine. It would be well when not satisfied with the performance of an engine to construct theoretical indicator cards and compare them with actual cards.

Steam Cards Showing Miscellaneous Troubles. From our study of the indicator diagram, it is evident that a great deal of useful information may be obtained by the correct interpretation of them. Fundamentally, the diagram is to register pressures for given piston positions, so all the information that is obtained in

addition to this, comes from a source of reasoning. A few cards, illustrating information which may be obtained, are given in Figs. 43 to 64, inclusive.

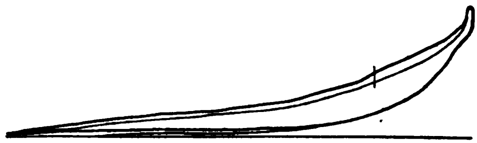


Fig. 43. Diagram Showing Improper Valve Lubrication

Valve Trouble. Figs. 43 and 44 illustrate cards taken from the h.e. of one of the cylinders of a locomotive running at thirty miles per hour, using 240 pounds steam pressure, with the reverse lever placed in the second notch ahead of the center position. This locomotive has a superheater, hence, with the high boiler pressure and superheat, trouble was experienced with the lubrication of the valves. This is indicated by the reduced area and distorted card shown in Fig. 43 as compared with that illustrated in Fig. 44. The card shown in Fig. 44 was obtained about twenty minutes later than that illustrated in Fig. 43. In Fig. 44 lubricating oil had been forced into the steam chest. The effect on the card shown in Fig. 43 was caused by the valve clinging to its seat, resulting in a shorter travel and poor steam distribution.

Fig. 44. Improvement in Diagram by Use of Lubricant

Fig. 45. Diagram Showing Sticky Indicator Piston

Sticky Indicator Piston. Figs. 45 and 46 show cards obtained from the c.e. of one of the cylinders of the same locomotive. The wavy appearance of the steam line in Fig. 45 is due to a dry, sticky

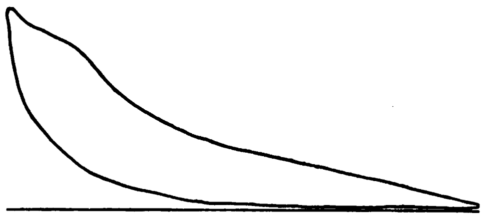


Fig. 46. Diagram After Trouble of Fig. 45 has been Removed

indicator piston. Fig. 46 illustrates the appearance of the steam line after the indicator piston had been removed, well oiled, and replaced.

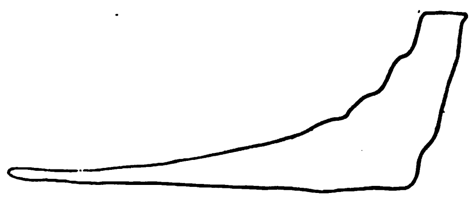


Fig. 47. Distorted Card Due to Binding Indicator Piston

Tight Indicator Piston. The cards exhibited in Figs. 47, 48, and 49 illustrate the distortion of the card which may occur when the indicator piston does not fit properly and binds, due to the indicator

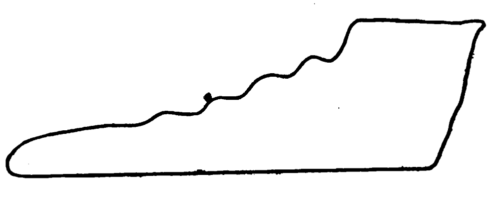


Fig. 48. Distorted Card Due to Binding Indicator Piston

parts not being put together properly. The indicator by means of which these diagrams were obtained had the screw in the bottom of the piston run up so far that the piston rod did not fit down over the projection on the piston, hence perfect alignment was not obtained.

Of the three cases, Fig. 49 is the worst. The area of the card is very much decreased and the back-pressure line is high.

Lost Motion. The effect of lost motion in the connections of an indicator is apparent in the cards illustrated in Figs. 50 and 51. It

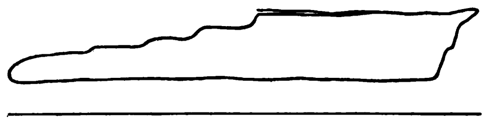


Fig. 49. Bad Case of Binding Indicator Piston

is, perhaps, most noticeable in the wave in the expansion line and the height of the back-pressure line.

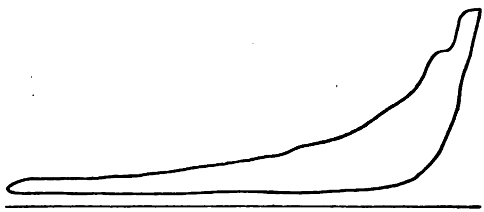


Fig. 50. Effect on Diagram of Lost Motion in Indicator Connections

Variable Cut-Off. In Fig. 52 is shown a card taken from a Buckeye engine at a speed below that at which the governor sets. With the engine working under this condition, the greatest c.o. is obtained.

Fig. 51. Diagram Showing Effect of Lost Motion in Indicator Connections

Fig. 53 illustrates a card taken from the same engine, running at 200 r.p.m. and with a load slightly under full load. Fig. 54 illustrates another card taken from the same engine operating under a very light load. The c.o. occurs very early. The small area of the card suggests the small amount of work being done in the cylinder.

Long Indicator Cord. Fig. 55 illustrates the effect of too long an indicator cord. Comparing this diagram with those shown in Figs. 52, 53, and 54 taken from the same engine but from the other end

Fig. 52. Card from Buckeye Engine at Low Speed

of the cylinder, the distortion becomes very apparent. Fig. 56 illustrates a distorted card from the same engine, its distortion being due to the cord slipping off from the sector of the reducing motion.

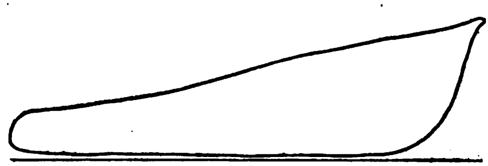


Fig. 53. Card from Buckeye Engine at Nearly Full Load

It is to be noted that there is a small loop in the top of this card which indicates too much compression. Oftentimes this loop appears when there is nothing wrong with the indicator or its attachments, but is an indication of disarrangement of the valves of the engine.

Fig. 54. Card from Buckeye Engine for Very Light Load

Speed Governing. There are two ways of governing the speed of an engine, namely, by throttling the steam or by varying the point of c.o. to suit the load conditions. The effect of these two methods on the indicator diagram is shown in Figs. 57 and 58. The diagram, Fig. 57, was taken when the speed was maintained constant by changing the point of c.o., this being decreased as the load decreased,

thus reducing the power in the cylinder. The card in Fig. 58 was obtained when the speed of the engine was maintained constant by throttling the steam supply rather than by changing the point of c.o.

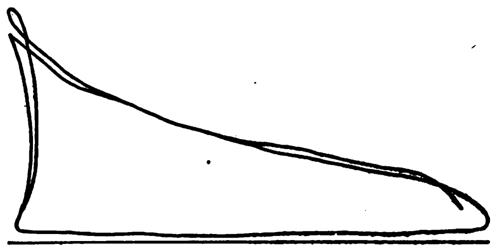


Fig. 55. Card Showing Effect of Long Indicator Cord

It should be noted that the area of the card is reduced in the same manner as when the point of c.o. was changed but that the events of the stroke remain unchanged under all conditions of throttling. This

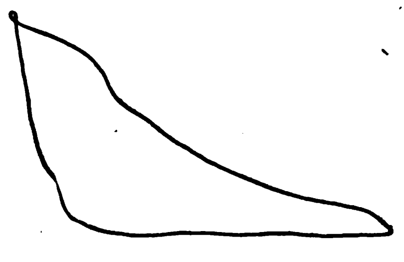


Fig. 56. Card Showing Slipping Indicator Cord

is not true, however, of the cut-off governor, because in this type, by changing the point of c.o., the other events of the stroke are affected in some degree.

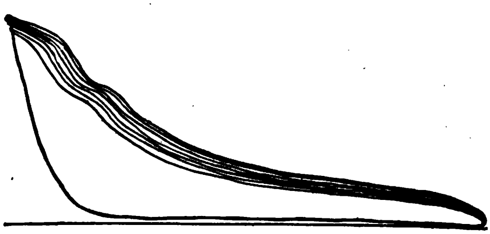


Fig. 57. Card Showing Effect of Changing Cut-Off

Faulty Valve Arrangement. A typical card taken from a Straight Line engine, running at 270 r.p.m. at full load, is shown in Fig. 59.

Aside from illustrating the form of card that this particular type of engine gives, it is of interest because it indicates a faulty valve arrangement. Referring to the figure, it will be seen that admission occurs

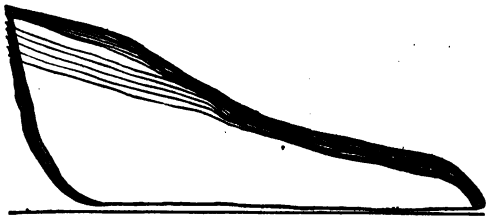


Fig. 58. Effect on Card by Throttling the Engine

at the end of the stroke as indicated at *a*. Late admission is indicated by the sloping admission line, giving the space *b* between the

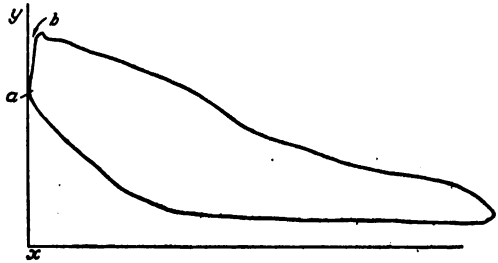


Fig. 59. Effect of Faulty Valve Arrangement

end of the stroke and the point where full admission occurs. Early admission would be indicated in the same way, the exception being

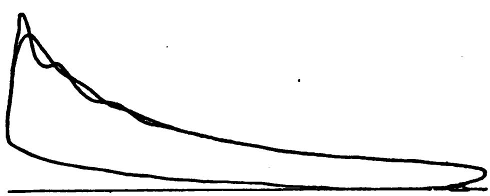


Fig. 60. Card of Gas Engine Operating Under Full Load

that the admission line would slope towards the line *x y* drawn at the end of the card, instead of away from it, as it does in the case illustrated. When retarded admission occurs in a very large degree, the curvature of the admission line is more pronounced.

Gas Engine Cards. It seems desirable to show a few typical gas engine cards, as every engineer may be called upon to indicate gas engines as well as steam engines. Fig. 60 is a diagram obtained from a gas engine operating under a full load of approximately 18 h.p. A 240-pound spring was used in taking the card. Fig. 61 is a diagram taken from the same engine, but operating under different conditions. It shows the change in the diagram produced by throttling the mixture for various loads. This card also shows that the indicator cord stretched slightly, otherwise the different maximum compression points *a* would have fallen on a line perpendicular to the atmosphere line.

In most of the diagrams presented thus far, the errors pointed out were those due chiefly to defects in the operation and attachment of the indicator, or in lubrication.

Fig. 61. Change in Gas Engine Diagram by Throttling the Mixture for Various Loads

Cards Showing Valve Troubles. It is now desired to direct attention directly to defects in valve-setting as shown by the indicator diagram, to the end that suggestions may be given as to how to properly adjust the valves of an engine by the use of an indicator.

The most common faults in the distribution of steam in an engine cylinder can be divided into four classes, viz, admission too early or too late; cut-off too early or too late; release too early or too late; and compression too early or too late.

Late Admission. The diagram, Fig. 59, shows too late admission, as was previously pointed out. If a plain slide valve were used, the reason why admission occurred too late was because the angle of advance was too small. If admission seems too early, the opposite thing is true and the angle of advance should be decreased.

Excessive Back Pressure. The cards shown in Figs. 47, 48, and 49 portray too much back pressure. While in these cards it was due to defects in the indicator, rather than in the engine, yet this excessive back pressure is sometimes found due to inherent defects in the

design of the engine, such as too small exhaust ports or pipes, or to the passage of steam through coils of pipe for heating purposes. Excessive back pressure is an indication of a loss of power and should be kept as small as possible. If the exhaust steam is used for some

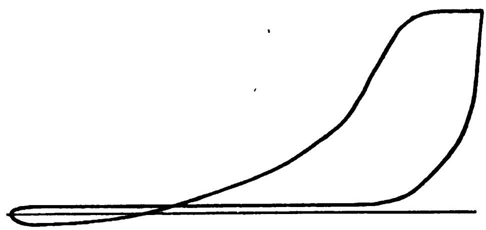


Fig. 62. Diagram Showing Effect of Too Early Cut-Off

useful work, such as heating, etc., an increased back pressure above the normal is permissible.

Early Cut-Off. The diagram, Fig. 62, shows the c.o. to come too early. In this case the c.o. is so early that the expansion line extends below the atmosphere line, making a loop. In finding the area of such a card for computing the power, the area of the loop must be subtracted from the total area. In using a planimeter to determine the area, it will automatically make the reduction so the reading will be correct. This loop is frequently spoken of as negative work.

Wire Drawing. Fig. 63 shows a pair of diagrams from a plain slide-valve engine. The admission lines are good. The sloping steam lines show wire drawing due to the slow action of the valve

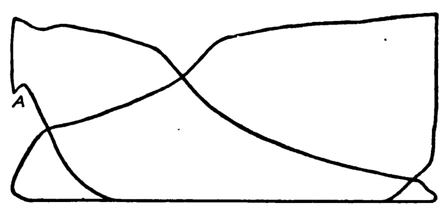


Fig. 63. Pair of Diagrams from Plain Slide-Valve Engine

or too small ports or pipes. This wire drawing decreases the area of the card and indicates a loss. The greatest fault is the inequality of area of the diagram. The late cut-off and consequent late compression of one end causes more area than the too early cut-off

and too early compression of the other end. These cards can be improved upon by adjusting the angle of advance of the eccentric and the length of the valve rod. If the left card were a normal one the hook at *A* might indicate an open cylinder cock.

Early Compression. The diagram of Fig. 64 indicates too early compression. The compression curve extends above the initial pressure line. The area of the loop must be subtracted from the card area when computing the i.h.p. If the cut-off is kept the same and the compression made what it should be, the gain in area would be the area included between the full line and the dotted line plus the area of the loop. The remedy for this case is to decrease the inside lap, which would permit exhaust to occur earlier and compression later.

The amount of compression an engine should have varies with the speed and type. Slow speed engines require less compression or

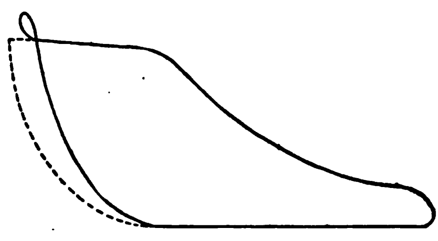


Fig. 64. Diagram Showing Early Compression

cushioning than high speed engines. The exhaust steam should never be compressed higher than the boiler pressure.

If the valve travel is increased, compression is retarded—that is, decreased—and release occurs sooner.

TESTING STEAM ENGINES

In the beginning of this study, it was stated that the indicator had been largely responsible for the refinement of the modern steam engine. In what way the indicator has influenced the development will be evident from the suggestions which follow and from the work involved in the testing of engines. The testing of steam engines requires considerable preliminary work and very careful attention to details. The tests may be made to ascertain whether the valves

are properly set; to determine i.h.p., b.h.p., and f.h.p.; to determine the amount of steam used per i.h.p. per hour, or the commercial efficiency; and to investigate the transference of heat between the steam and the cylinder walls, and losses due to this transference. It should be borne in mind that most of the results sought for are closely allied, so that one complete test may give data sufficient to obtain the value of all the factors mentioned. For instance, if one is seeking the loss due to friction, he must obtain the b.h.p. and i.h.p. and, having these, it is an easy matter to obtain the mechanical efficiency.

Factors Considered. Usually the principal object in testing a steam engine is to determine the cost of power or the effect of such conditions as superheating, jacketing, varying the point of cut-off, varying the point of compression, clearance, steam pressure, etc., upon the steam economy of the engine. We must determine, therefore, first, the cost of fuel, and second, the actual amount of heat used. In either case, the horsepower of the engine must be determined.

The indicated power is determined by means of the indicator, and the actual power delivered, by means of a dynamometer or friction brake. To determine the cost of power in terms of coal, it is necessary to conduct a careful boiler test, usually of twenty-four hours duration.

When the cost is expressed in terms of steam per horsepower per hour, we may follow either of two methods, viz, we may condense and weigh the exhaust steam, or we may weigh the feed water supplied to the boiler. When the object of the test is primarily for an investigation of the performance of the engine, it is best to weigh the condensed steam. This is the method used in the test described herein. An hour under favorable conditions is usually sufficient for such tests. Steam used for any purpose other than running the engine must be determined separately and allowed for.

Probably the most accurate terms in which to state the performance of an engine is in B.T.U. per horsepower per minute. When the cost is expressed thus, it is necessary to measure the steam pressure, amount of moisture in the steam, and temperature of condensed steam when it leaves the condenser. Jacket steam must be accounted for separately. Engines with their boilers, etc., for large plants, are usually built under contract to give a certain efficiency, and their

fulfillment of this contract can be determined only by a complete test of the entire plant. Before beginning the test, the engine should be run for a sufficient length of time in order to limber it up and get it thoroughly warmed. It is of the utmost importance that all conditions of the test should remain constant, especially the boiler pressure and the load. All instruments used in the test should be calibrated before being used, in order to determine the effect of any errors to which they may be subject.

Thermometers. All important temperatures, such as feed water, injection water, condensed steam, etc., must be taken by accurate thermometers, the errors of which have been previously determined and allowed for. Good thermometers sold by reliable dealers are usually satisfactory. Cheap thermometers are of little value in an engine test.

Indicators. The most important and in many respects the least satisfactory instrument used in the test is the indicator. It is subject to an error of 2 to 3 per cent, depending on the conditions. It does not work satisfactorily at more than 400 revolutions per minute. If the indicator is carefully tested under conditions similar to those under which it is used, the errors may be reduced to a minimum, but there will always be some uncertainty. The principal errors to which the indicator is subject have already been mentioned.

Scales. Weighing should be done on standard platform scales. The water may be weighed in barrels provided with large quick-acting drain valves which will allow the water to run out quickly. It is seldom possible to drain barrels completely, and so it is best to let out what will run freely, then shut the valve and weigh the barrel. This we call "empty" weight, and deducted from the weight "full" evidently gives us the true weight of water.

If not convenient to weigh the water, it may be measured in tanks or receptacles of known capacity, and the temperature taken, allowing the proper weight per cubic foot for water at that temperature; or it may be determined by meters.

Meters. Water meters are of two kinds, viz, those that record the amount of water by displacement of a piston, and those in which the flow is recorded by means of a rotating disk. Piston water meters can be made very accurate, and if working under fair conditions of service, they may be relied upon to a close degree. The

chief error in a meter arises from the air that may be in the water. To reduce this error to a minimum, the meter should be vented so as to allow the air to escape without passing through the meter. Rotary meters are good enough for very rough work, but are seldom sufficiently accurate for a careful engine test. So far as possible weirs should not be used in engine work. They may be fairly accurate under certain conditions, but a very little oil in the water may affect them seriously. They may sometimes be used to measure the discharge from a jet condenser, for then the volume is so large that the actual error is proportionately small. The use of meters for testing purposes should always be discouraged. When used, however, they should always be carefully calibrated under as nearly as possible the same conditions as existed during the test.

Gauges. Pressures should be measured on good gauges that have been recently tested. The atmospheric pressure should be read from the barometer, and for accurate work this pressure should be used. For ordinary work, 30 inches of mercury, or 14.7 pounds, may be used.

Calorimeters. When using superheated steam, it is sufficient to take the temperature and pressure in the steam pipe, but if saturated steam is used, we must determine the amount of moisture it contains. This is done by means of a calorimeter such as has previously been described.

Prony Brakes. Any of the forms of friction brake described will answer the purpose. For smooth and continuous running, it is essential that the brake and its band be cooled by means of water and that some lubrication be applied to the surface of the brake wheel. The water may either circulate in the rim of the wheel or around the brake band, but it must not come in contact with the rubbing surfaces.

Original Type. The most common form of brake used is some modification of the Prony brake as illustrated in Fig. 65. This is one of the simplest forms of absorption dynamometer. The two wood blocks *A* and *C* are held together against the rim of the pulley *P* by bolts. The thumb nuts, *E E*, are used to adjust the pressure. By means of the bolts, the arm *L* is held to the upper block. From this arm is suspended the ball weight *B* which, by sliding along the arm, counterbalances the weight of the arm and pan at the other end. The pulley revolves at the required speed in

the direction indicated by the arrow. The bolts are tightened until the lever remains stationary in a horizontal position when a known weight W is hung at the end. Suitable stops must be arranged at

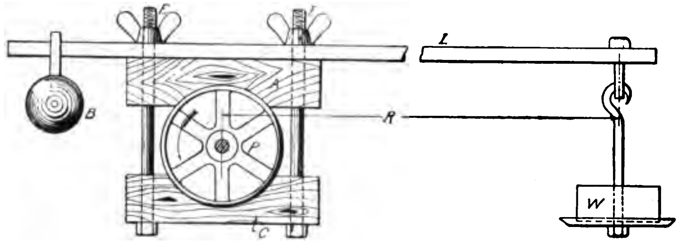


Fig. 65. Original Form of Prony Brake

the outer end of lever L to prevent an accident in case the brake should happen to grip the wheel and cause the weight W to be thrown over.

The amount of work absorbed by the brake depends upon the weight W, the length R, and the speed. It is independent of the diameter of the pulley and the pressure of the blocks because the moments of forces about the center of the pulley are equal when the

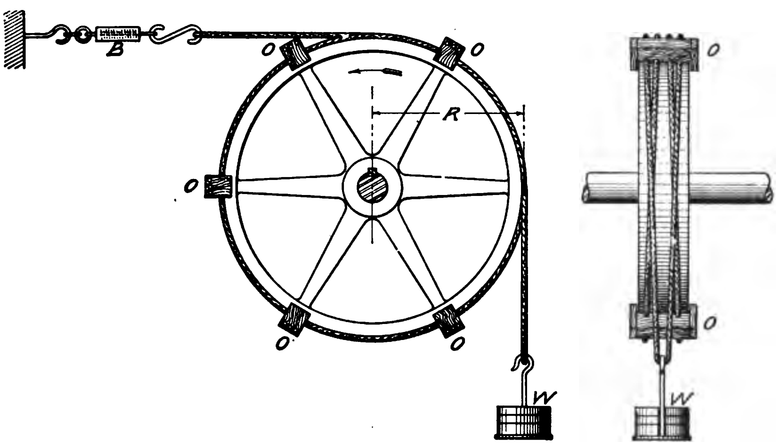


Fig. 66. Rope Form of Prony Brake

lever L is horizontal. Letting f equal the coefficient of friction, p the pressure of the blocks, and r the radius of the pulley, we have

$$f\,p\,r = W\,R$$

The work done at the face of the pulley equals the tangential force between the block and the wheel multiplied by the distance passed over, which also equals weight W multiplied by the number of feet W would move through if it were free to rotate.

Let N be the number of revolutions per minute. Then the distance passed through per minute equals $2\pi rN$ and the work done equals $2\pi rNfp$. Then as $fpr = WR$, the work done at the rim of the pulley equals the left-hand side of the equation multiplied by $2\pi N$, and to keep both sides equal we multiply WR by $2\pi N$. Then the work done per minute is obtained from the expression $2\pi NWR$.

$$\text{b.h.p.} = \frac{2\pi NWR}{33000}$$
$$= .0001904\ NWR$$

EXAMPLE. A Prony brake having an arm 4 feet long attached to the pulley of an engine sustains a weight in the scale pan of 50 pounds when the speed of the engine is 300 r.p.m. Find the brake horsepower.

$$\text{b.h.p.} = .0001904 \times 300 \times 50 \times 4$$
$$= 11.424$$

Rope Type. The rope brake shown in Fig. 66 is easily constructed of material at hand and being self-adjusting needs no accurate fitting. For large powers, the number of ropes may be increased. It is considered a most convenient and reliable brake. In Fig. 66 the spring balance B is shown in a horizontal position. This is not at all necessary; if convenient the vertical position may be used. The ropes are held to the pulley or flywheel face by blocks of wood O. The weights at W may be replaced by a spring balance if desirable.

To calculate the brake horsepower, subtract the pull registered by the spring balance B from the load at W. The lever arm R is the radius of the pulley plus one-half the diameter of the rope. The formula for power absorbed is

$$\text{b.h.p.} = \frac{2\pi RN(W-B)}{33000}$$

$$= .0001904\ RN\ (W-B)$$

If B is greater than W, the engine is running in the opposite direction. In this case the formula becomes

$$\text{b.h.p.} = .0001904\ R\ N\ (B-W)$$

Example. A rope brake is attached to a gas engine brake wheel. The average reading of the spring balance is 8 pounds when W is 80 pounds. If the radius of the brake wheel is 28 inches and the rope 1 inch in diameter, what is the b.h.p. when the engine makes 350 revolutions per minute?

$$R = 28 + \tfrac{1}{2} = 28\tfrac{1}{2} \text{ inches} = \frac{28.5}{12} \text{ feet}$$

Then from the equation for brake horsepower, we have

$$\begin{aligned} \text{b.h.p} &= .0001904\ R\ N\ (W-B) \\ &= .0001904 \times \frac{28.5}{12} \times 72 \times 350 \\ &= 11.4 \end{aligned}$$

If both the indicated horsepower and the brake horsepower are known, the power lost in friction may be found by subtracting the b.h.p. from the i.h.p.

Modern Band Type. The two forms of brakes shown in Figs. 65 and 66 serve their purpose very well but are not very durable.

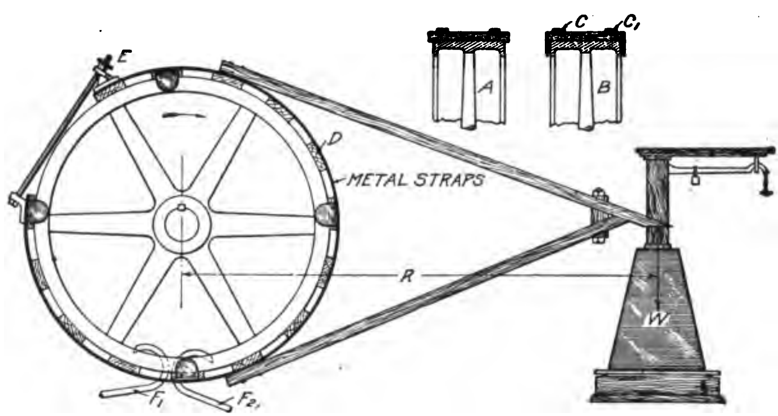


Fig. 67. Modern Band Form of Prony Brake

When it is desired to make repeated tests of an engine for a considerable period of time, or when it is desired to keep the machine in readiness for tests at all times, as in experimental engineering laboratories, it is better to provide a brake of the form illustrated in Fig.

67. This brake is made up of two metal straps CC_1, as shown in the cross-sectional view. Attached to these metal straps are a number of wood blocks placed at regular intervals. These blocks are made of hardwood and form the rubbing medium of the brake. The brake band and blocks are held in place on the pulley by having metal clips extending down the side of the pulley for a fractional part of an inch. The brake is tightened up by means of the hand wheel E. The pipe F_1 delivers water to the rim of the wheel for keeping it cool. Pipe F_2 is arranged to scoop up the water from the rim, thus keeping the rim of the wheel filled with cool water. Ordinarily pipe F_2 is not needed. The water in the rim will never be heated above the boiling point and this temperature will do no harm. When the engine is running in the direction indicated by

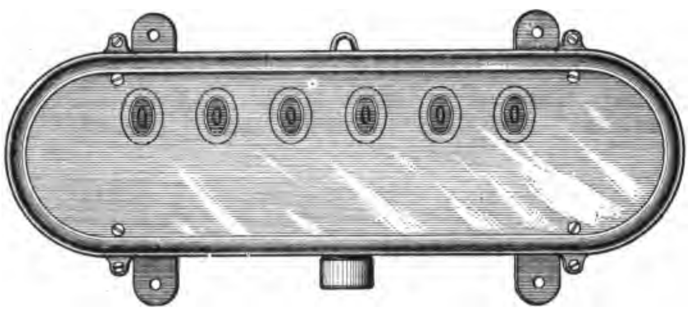


Fig. 68. Revolution Counter and Recorder

the arrow, the tendency is for the brake band to move in the same direction, but the V-shaped arms resting upon the platform scales prevent this, and the amount of pressure W exerted by the brake lever is weighed by the scales. Hence, one can at any time easily determine the work delivered to the brake. This form of brake is shown in application on a Buckeye engine in Fig. 10; the scales are not used, but instead the brake arm is connected to a chain, which runs over a quadrant to which it is attached. Attached to this quadrant is an arm that carries a weight B and a pointer E. The pointer indicates the pounds pull on the graduated arc C. By careful calibration, the arc C is graduated in pounds. In the brake shown in Fig. 67, the pressure W on the scale must be corrected before using the brake horsepower formula, for the unbalanced weight of the brake arm. If the brake band is supported on a knife edge imme-

diately above the center of the engine shaft, and the outer end of the shaft then weighs W_1 pounds, the brake horsepower formula would be

$$\text{b.h.p.} = .0001904\ RN\,(W - W_1)$$

Speed Counter. In finding the b.h.p. or i.h.p. of an engine, it is necessary to know the number of revolutions the engine makes in a minute. This speed is usually designated as r.p.m. In order to obtain the correct r.p.m., an instrument known as a revolution counter is usually attached to some revolving or reciprocating part of the engine. A common form of such a counter is shown in Fig. 68.

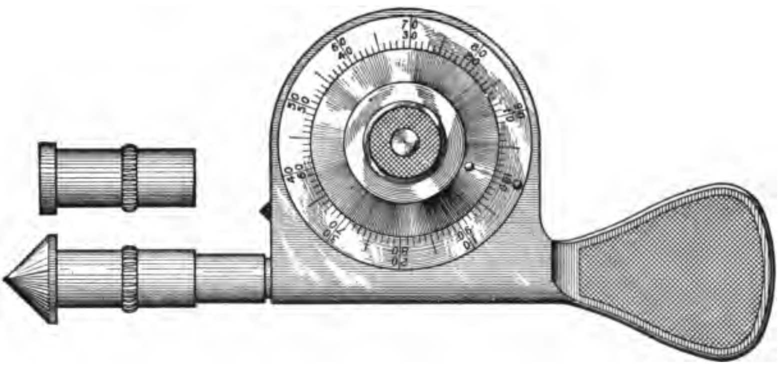


Fig. 69. Standard Form of Speed Counter

The actuating motion of the engine or other machine to which the counter is to be attached, is generally communicated by a rod or bar moving in the same general direction of its length, and the lever should be connected to such rod at a right angle when such rod is in the middle of its movement. It should not be clamped rigidly to the shaft until the latter is turned so as to bring the pawl to the middle of the stroke. It may be determined, practically, by opening the lid of the counter and watching the movement of the pawl as the shaft is rotated, when the middle point of its travel can be easily fixed. When in this position, clamp the crank to the shaft by means of the set screw.

This arrangement provides for the utilization of the entire motion of the actuating rod at the angle of greatest effectiveness in moving the mechanism of the counter; and if for any reason the

movement of the rod is shorter than its longest possible stroke—as might happen in the case of a direct acting pump—there would still be ample motion to insure a correct count.

This counter is adapted to either right or left rotary or reciprocating motions and is capable of 500 revolutions per minute with safety to the machine and accuracy in the enumeration.

The shaft through which the actuating force is applied may extend from the counter either on the right-hand or left-hand side, as desired.

A very simple form of speed counter is illustrated in Fig. 69. It has a rubber tip which is held in the center of the engine shaft. The motion of the engine shaft is transmitted to the shaft of the counter which drives through a system of gears a pointer, the latter indicating on a graduated dial the number of revolutions made in a given time. When well made, this is a very accurate instrument and may be read with reliability for speeds up to as high as two or three thousand r.p.m.

INDICATOR TROUBLES AND REMEDIES

Necessity for Care in Using Indicator. The steam and gas engine indicator is an extremely valuable instrument for engineering purposes when used intelligently, but when in the hands of a careless inexperienced operator the results obtained may be little short of worthless. The instruments constructed by reputable manufacturers are reliable for the purposes for which they were intended and are indispensable in a steam or gas engine power plant of any considerable size. For scientific and investigative purposes the most reliable instrument should be used and the operator should be careful and experienced, in order that the best possible results may be obtained. Many operators make use of the indicator with a desire to secure reliable information and in many instances are sincere and painstaking in their efforts, but, unless they give proper attention to certain fundamental precautionary rules, the accuracy of the results secured may be questioned.

Attachment of Indicator. *Short Connections Desirable.* As has been previously stated, in order to secure reliable diagrams, the indicator should be attached as close to the cylinder as conditions of the particular installation will permit. Long pipe connec-

tions result in unreliable indications. Generally speaking, other conditions remaining the same, the shorter the connections the more accurate the results. Most modern steam engines are now made with suitable holes which are tapped for indicator connections. When the cylinders are not drilled and properly tapped for receiving the indicator, the engineer in charge should be competent to do it under the directions here given.

Conditions Affecting Location of Indicator. Before deciding just where the holes should be drilled, it is desirable that all conditions of the case be carefully studied with a view of devising the whole plan for indicating the engine. It usually happens that the reducing motion, or drum motion as it is sometimes called, can be erected more advantageously in one position relative to the engine than another, or one kind may be better adapted for a given place than another. The type of engine, location of the steam chest or valves, the kind of cross-head and the best means of attaching to it, and the position of the eccentric, its rods, and connections, all should be given careful consideration in determining the best places to locate the indicator. A free passage for steam to the indicator is of prime necessity and a location of the indicator insuring convenience in operation is desirable. The instrument can be used in a horizontal position but in taking diagrams it is more convenient when in a vertical position. Then again, the vertical position is that in which it would most probably be calibrated and for this reason alone is preferable. A prominent manufacturer gives the following directions for drilling cylinders to receive indicators:

Mounting Indicator on Cylinder. When drilling holes in the cylinder the heads should be removed if convenient, so that one may know the exact position and the size of the ports and passages and be able to remove every chip or particle of grit which might otherwise do harm in the cylinder or be carried into the indicator and injure it. When the heads cannot be taken off, it can be arranged so that a little steam may be let into the cylinder when the drill has nearly penetrated its shell, so that the chips may be blown outward—care being taken not to scald the operator.

It is essential that the holes be drilled into the clearance space at points beyond the travel of the piston so as not in any way to obstruct the passage of steam to the indicator. The most common practice in the case of horizontal engines is to drill and tap the holes in the side of the cylinder at each end. On certain types of horizontal engines, it is possible to drill and tap into the top of the cylinder at each end, in which case the indicator cocks can

be screwed directly into the holes. On vertical engines, the upper indicator is frequently connected into the cylinder head, although better results will be obtained if both holes are drilled and tapped in the side of the cylinder.

Reducing Motions. It sometimes happens that in an effort to get quick results a makeshift type of reducing motion, or drum motion, is resorted to, with the almost inevitable result that the diagrams secured by its use are extremely faulty and in some cases worthless. In the long run, the most satisfactory results are secured if some form of approved reducing motion, such as has already been described, is used. Results of experience have shown that diagrams varying in lengths from $2\frac{1}{2}$ to $3\frac{1}{2}$ inches, depending upon the speed of the engine, are most satisfactory. These lengths have been found long enough to admit of all useful divisions, and the movement of the card is slower and the tracing correspondingly more delicate and accurate than if a longer card is made. These facts should be borne in mind in designing and proportioning the reducing motion.

Drum Spring Tension. A great many operators give no attention to the tension of the drum spring, using the same adjustment for testing engines operating at wide ranges of speeds. Theoretically speaking, there is only one correct drum spring tension for one speed, other conditions remaining unchanged, but the refinement need not be carried to this point. However, it is a matter which should at least receive some attention. For a particular installation and speed the tension should be a sufficient amount, and no more, to overcome the friction of the pencil on the paper and maintain, at all times, the indicator cord taut. Any very great amount of tension, in addition to that necessary, not only affects the wearing qualities of the instrument but shortens the life of the indicator cord.

Adjustment of Guide Pulley. As has been heretofore explained, the object of the guide pulley is to properly conduct the indicator cord from the drum to the reducing motion. It is such an insignificant piece of mechanism that it is frequently overlooked by the inexperienced operator. Whenever this occurs, the diagrams are usually unsatisfactory in many respects, as can readily be seen, and in a very short time there results a broken indicator cord which, under certain conditions, is extremely

difficult to repair. The adjustment of the guide pulley is one of the first adjustments which should be made in setting the indicator for taking cards.

Adjustment of Pencil Pressure. In the early forms of indicators, the diagram was drawn on plain paper by means of a graphite pencil, the pencil being sharpened to a fine round point by means of a knife or fine file. The graphite pencil can still be used but a more satisfactory result is obtained by the use of a brass point for a pencil in connection with chemically prepared paper, known as metallic paper. In either case the result desired is the securing of a light distinct diagram, that is, a diagram which is distinct yet made by using a pencil pressure no greater than is absolutely necessary. This is a matter that is quite generally overlooked by the average operator, the tendency being to obtain a diagram showing much contrast. The operator should always remember that within certain limits the lighter the line the more accurate the results. If too great a pressure is employed between the pencil and paper, the pencil will lag on both ascending and descending pressures, with the result that the diagram will be too small and will not represent the true power of the engine. It will not only give an incorrect indication of the power and pressures but also will improperly represent the true location of the various events of the cycle.

Miscellaneous Precautions. *Importance of Rules for Use of Indicator.* It should be the sincere effort of every operator to secure the very best results possible when making use of the indicator. To this end very careful attention should be given to the brief rules prepared by the manufacturers for assembling and manipulating the instrument. At least attention should be given to these general directions until one becomes thoroughly familiar with the apparatus. For example, in the case of the Crosby indicator, the directions given on pages 30 to 33 of this text, inclusive, should be thoroughly digested and mastered. No matter where the indicator is made or by whom, the operator should adopt a correct and regular method of procedure in its use so that it becomes a habit.

Care in Handling Indicator. Perhaps one of the chief reasons that the indicator receives so many damaging knocks and blows is

the manner in which it is removed from its carrying case and assembled. On opening the carrying case preparatory to taking diagrams, the indicator should at once be lifted out and attached to the indicator cock where it will be securely held while the spring and parts are being connected and adjusted. Before attaching the indicator, however, the cock should be opened for a very brief time and steam be permitted to blow through so as to blow out any foreign matter which might be detrimental to the correct action of the instrument. When the indicator is not in use, it is preferable to place a cap on the indicator cock, which cap is usually furnished with the indicator.

In lifting the indicator from one position to another, never do so by taking hold of the drum as many instruments are rendered useless by carelessness in this regard. In some designs the drum is not held in position by means of a small thumb nut but slips off very easily.

Lubrication. The question of proper lubrication is one which should be handled intelligently. Always before placing the piston and spring in the proper working position, the piston should receive a generous supply of oil. For steam engine work a good grade of valve oil should be employed, while for gas engine work a good quality of gas engine oil should be used; machine oil should never be used. For air compressor work and hydraulic work a high grade of light oil should be used. Occasionally the pencil mechanism should receive oil, which should be light and offer little tendency to cause gumming. Cases are on record in which diagrams were taken with the indicator piston lubricated with the wrong kind of oil and much time was spent and trouble experienced in an effort to diagnose an apparent error which did not exist. Hence the necessity for proper lubrication.

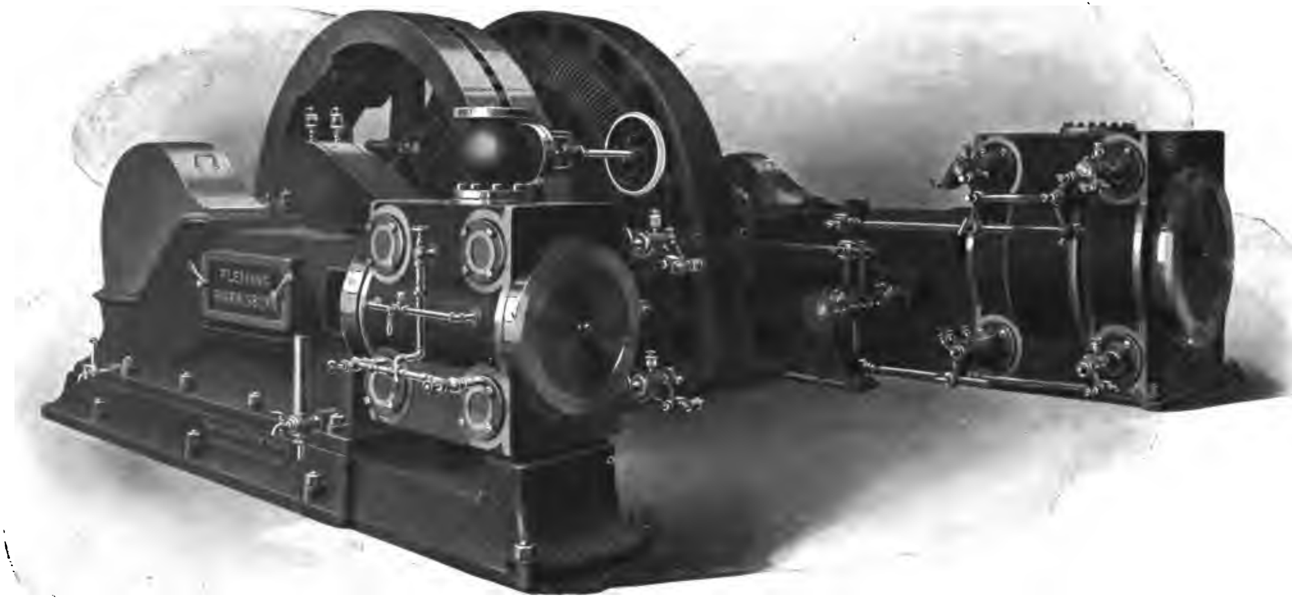
Causes of Incorrect Indication. In taking diagrams many things may happen which will result in incorrect indication. In taking a series of cards, if one notices a card which is much shorter than all others, it may be due to one of two things: either the indicator cord is not properly connected to the reducing motion or by some means it has become too long.

It will sometimes happen that the pressures on one card are much smaller than on others in the series. When this occurs, if

there has been no material change in steam pressure, it is very probably caused by the indicator cock being opened only partially. A leaky indicator cock is always a source of much annoyance. It not only causes incorrect indications of pressures, especially on the exhaust side of the diagram, but produces an irregular atmospheric line which otherwise would be straight.

A diagram which shows an abnormal back pressure, when the engine is known to have but very little back pressure, is most probably due to a loosely connected piston or pencil mechanism, unless it is caused by a sticky piston. In the latter case, however, other indications on the diagram would probably reveal the facts.

Modifications of Indicator for High Speeds. The indicator as usually constructed will give satisfactory results for all ordinary speeds. It cannot be used successfully for speeds above 450 revolutions per minute. For the higher speeds it is necessary to use a heavier spring than would be needed for the same pressure at lower speeds. For such high speeds it is also desirable to use a reducing motion proportioned so as to give a card having a length not to exceed 2 or 2½ inches. Best results are secured when the indicator is used at speeds below 200 revolutions per minute.



FLEMING-HARRISBURG CROSS-COMPOUND HEAVY-DUTY CORLISS VALVE ENGINE
Courtesy of Harrisburg Foundry and Machine Company, Harrisburg, Pennsylvania

PART II

VALVE GEARS

VALVE CHARACTERISTICS

Function. Steam enters the cylinder of a steam engine through ports which must, in some manner, be opened and closed alternately, in order to admit and exhaust the steam at the proper time. To accomplish this purpose, one or more valves are moved back and forth across the port openings. A complete understanding of the valve and valve gear is essential to the engineer as well as to the designer, for even though a valve be properly designed, the economy of the engine may be seriously impaired by improper valve setting. The design and adjustment of these valves play a very important part in the efficient action of the steam engine.

A valve gear is a mechanism consisting of a combination of slotted links, eccentrics, rods, levers, and other devices designed to operate valves of various types. The valve gear is separate and distinct from the valve. It operates the valve or valves but, strictly speaking, is not a part of them. This being true, one type of valve gear may be applied or used in connection with several different types of valves. For instance, the Stephenson gear may be used to operate a plain slide valve on one engine, a piston valve having either inside or outside admission on another, while a third may be attached to a more complicated form of valve mechanism. It should be borne in mind, therefore, that the valve gear is a separate and distinct part of the steam engine and that its function is to impart motion to the valve or valves.

The valves, in turn, perform the following functions during the engine cycle:

(1) *Admission.* This begins when the valve opens to admit steam to the cylinder.

(2) *Cut-Off.* This is the point at which the valve closes to cut off the admission of steam.

(3) *Expansion.* This takes place from cut-off to release.

(4) *Release.* This begins when the exhaust port is opened.

(5) *Compression.* This begins when the exhaust port is closed.

There may be a single valve to regulate admission and exhaust or there may be a double set of valves, one set to admit the steam at each end and another to release it. The valve may have a plain reciprocating motion, moved either by a rod or by some device that releases at the proper time, allowing the port to close suddenly under the influence of counterweights, springs, or vacuum dashpots. To the first class belong the plain slide valve and its modification of piston valve, gridiron valve, etc.; to the second class belong such valves as the Corliss, the Brown, and others.

The simplest type of valve is the plain slide, or *D*, valve as shown in Fig. 1, in which *V* is the valve, *R* is the valve rod, *K* is the exhaust cavity, P_1 and P_2 are the steam ports, *E* is the exhaust port, *A B* is the valve seat, and *D M* are the bridges of the valve seat. The valve seat must be planed perfectly smooth, so that steam pressure on the valve will make a steam-tight fit and cause as little friction as possible when the valve moves back and forth. Furthermore, the length of the seat *A B* must be a little less than the distance from the extreme right-hand position of the right-hand edge of the valve to the extreme left-hand position of the left-hand edge of the valve. This allows the valve at each stroke to slightly overtravel the seat, thus keeping it always worn perfectly flat and smooth. If the valve seat were not raised slightly above the rest of the casting, or if it were too short, the constant motion of the valve would soon wear a hollow path in the valve seat, and it would cease to be steam tight.

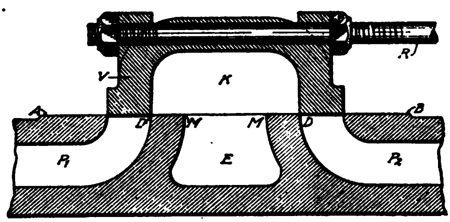


Fig. 1. Plain Slide, or D, valve

Eccentric. The valve usually receives its motion from an eccentric, which is simply a disk keyed to the shaft in such a manner that the center of the disk and the center of the shaft do not coincide. It is evident that as the shaft revolves, the center of this eccentric disk moves in a circle about the shaft as a center, just as if it were at the end of a crank. The action of the eccentric is equiva-

lent to the action of a crank whose length is equal to the distance between the center of the eccentric and that of the shaft.

Fig. 2 represents the essentials of an ordinary eccentric. O_1 is the center of the shaft, O_2 is the center of the eccentric disk E, and S is a collar encircling the eccentric and attached to the valve rod R. As the eccentric turns in the strap, the point O_2 moves in the dotted circle around O_1 and the point A_1 also moves in a circle. When half a revolution is accomplished, the point O_2 will be at O_3, the point A_1 will be at A_2, and the eccentric strap and valve rod will be in the position indicated by the dotted lines. The distance O_1O_2 of the

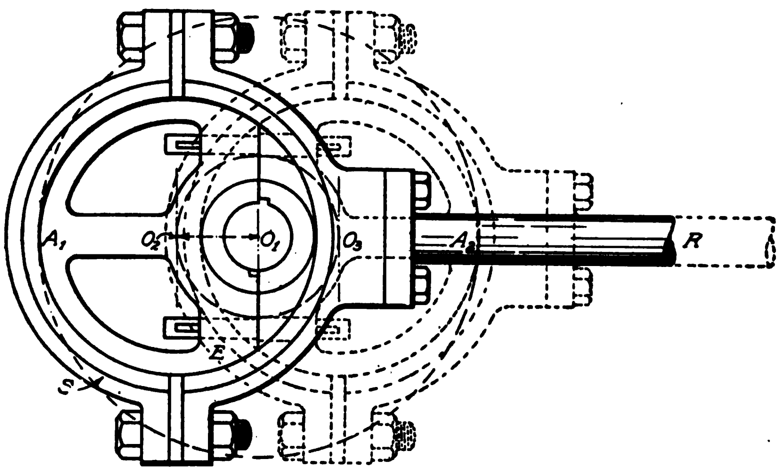


Fig. 2. Details of Ordinary Eccentric

center of the eccentric from the center of the shaft is known as the *eccentricity*, or *throw*, of the eccentric. The *travel* of the valve is twice the eccentricity.

Since the eccentric transmits the motion of the revolving shaft to the valve, it will be necessary to study the relative motions of crank and eccentric in order to obtain a clear idea of the steam distribution. This relation will be developed in connection with the discussion of the valve action which follows.

Valve Motion. *Valve without Lap.* Fig. 3 shows a section through the steam and exhaust ports of an engine, together with a plain slide valve placed in mid-position* and so constructed that

* A valve is in mid-position when the center line of the valve coincides with the center line of the exhaust port.

in this position it just covers the steam ports. Referring to Fig. 1, which shows the same type of valve drawn to a larger scale, suppose the valve is moved a slight distance to the right; the port P_1 is then

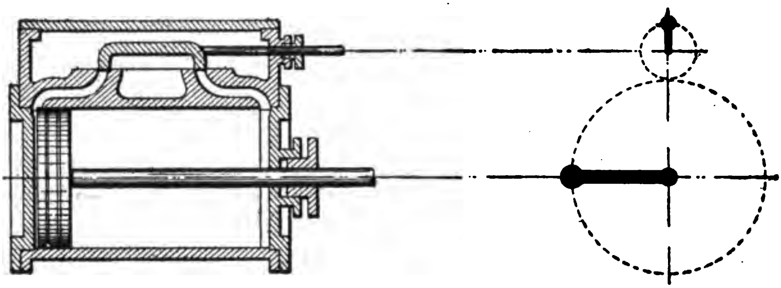


Fig. 3. Cylinder Details Showing Plain Slide Valve without Lap in Mid-Position

uncovered and opened to the live steam which enters the cylinder and causes the piston to move. Since the two faces of the valve are just sufficient to cover the steam ports, it is evident that as the port P_1 opens to live steam, the port P_2 opens to the exhaust. The ports are closed only when the valve is in mid-position. This allows admission and exhaust to continue during the whole stroke. With such a valve, there is no expansion or compression; the indicator card is a rectangle, and the m. e. p. is equal to the initial steam pressure, assuming no frictional losses in the steam pipe or condensation in the cylinder.

For a theoretical discussion of valve motion, it is assumed that the eccentric rod moves back and forth in a line parallel to the center

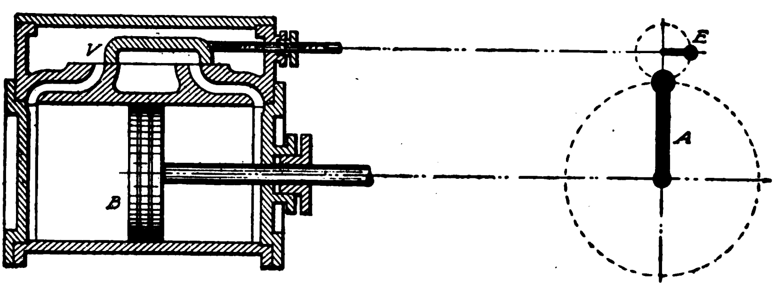


Fig. 4. Position of Piston and Valve in Cylinder Shown in Fig. 3, after One-Half Stroke

line of the engine. This is not the case in practice, for the eccentric rod always makes a small angle with the center line, just as the connecting rod does, but the eccentricity is so small in comparison with

the length of the eccentric rod that the angularity of the eccentric rod is very much less than the angularity of the connecting rod, and its influence may be neglected without appreciable error.

When the valve shown in Fig. 3 is in mid-position, the crank is on dead center, the eccentric is set at right angles to it, and the piston is just ready to begin the stroke.

Fig. 4 shows the relative positions of the crank *A*, piston *B*, eccentric *E*, and valve *V*, when the crank has made a quarter turn or the piston has moved to about half-stroke. The eccentric is now in its extreme position to the right, the valve has its maximum dis-

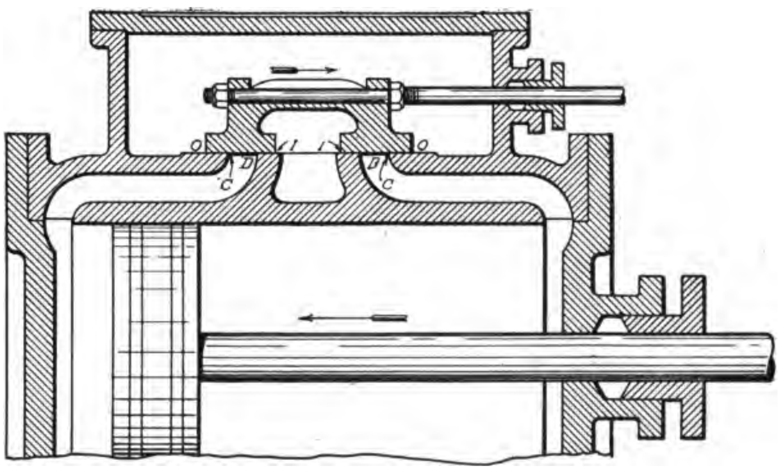


Fig. 5. Details of Cylinder, Showing Valve with Lap

placement, and both the steam and exhaust ports are wide open. The valve will not close again until the piston has reached the end of its stroke.

This type of valve is used only on small engines and, since it allows no expansion of the steam, is very uneconomical. Furthermore, it will be seen that this valve opens just after the stroke begins, which is impractical, for it means that the piston has begun its stroke before the full steam pressure reaches it, which will cause an inclined admission line on the indicator diagram.

Valve with Lap. If the face of the valve is made longer than shown in Fig. 1, so that in mid-position it overlaps the steam ports, we shall have a valve such as shown in Fig. 5. The amount that

the valve overlaps the steam ports when in mid-position is called the *lap* of the valve. In Fig. 5, *D I* is the *inside lap* and *O C* is the *outside lap.*

It will at once be seen that both the admission and exhaust ports may remain closed during a part of the stroke, thus making expan-

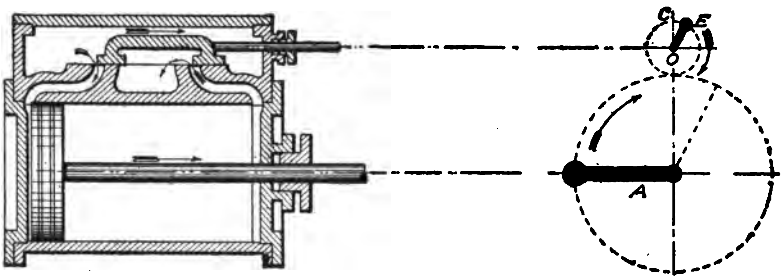


Fig. 6. Valve with Inside and Outside Lap Set for Admission

sion and compression possible. It is also evident that steam can not be admitted until the valve uncovers the port by moving from mid-position a distance equal to *O C*. Admission continues until the valve returns to such a position that the outer edge of the valve again closes the port. Release will begin when the inner edge of the valve begins to uncover the port.

Analysis of Motion. Fig. 6 represents a valve, having both inside and outside lap, which is set at the point of admission. Since

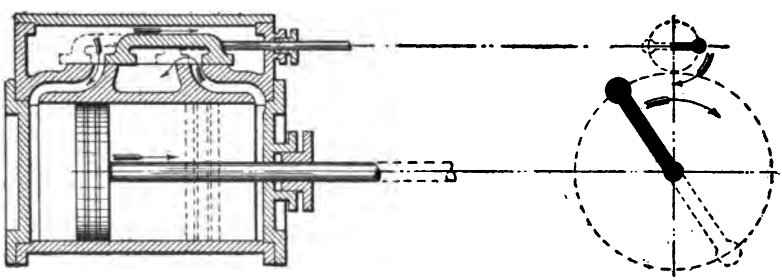


Fig. 7. Valve Set at Maximum Displacement

the valve must move over a distance equal to the outside lap in order that admission may take place under proper conditions, it is evident that the eccentric can no longer be at right angles to the crank at the beginning of the stroke, but must be in advance of the

right-angle point by an amount equal to the angle EOC, known as the *angular advance.*

The maximum displacement of the valve is attained when the eccentric is horizontal, as shown in Fig. 7. In this position, both the steam and the exhaust ports are wide open, and any further

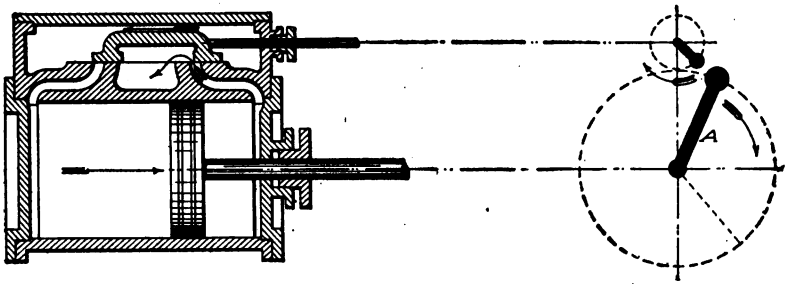


Fig. 8. Valve Position with Steam Port Closed on Head End

motion of the piston will cause the valve to move toward its mid-position.

Admission continues until the valve returns to the position shown in Fig. 8. Here the outside lap just closes the left-hand steam port, cut-off takes place, and the steam already in the cylinder begins to expand. As the valve continues to move toward the left, the left-hand inside lap begins to uncover the left-hand port and releases the steam at the position shown in Fig. 10. The dotted lines of Fig. 7 show the valve in its extreme position to the left, while the

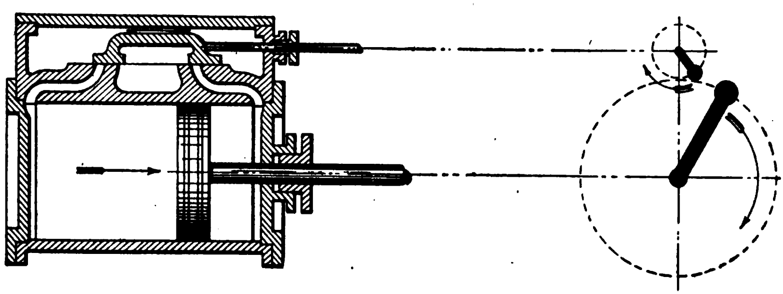


Fig. 9. Valve Position with Exhaust Port Closed on Crank End

dotted position of crank and eccentric in Fig. 10 shows the valve returned to the point of compression, which continues until the conditions of Fig. 6 are again reached and the opening valve allows steam again to enter the cylinder.

This process has been traced step by step for one end only; let us now consider what is happening at the other end. While the crank A is moving from the position shown in Fig. 6 to that in Fig. 8, steam is being admitted to the head end and being exhausted from

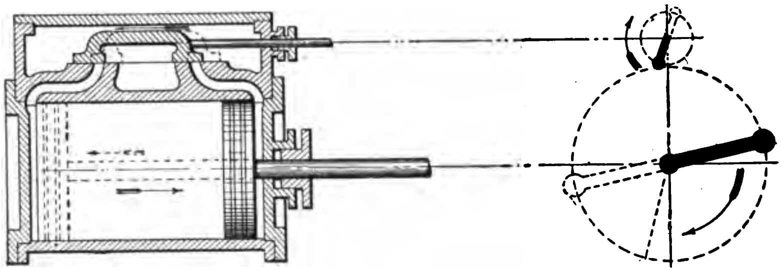


Fig. 10. Position of Valve and Cylinder for Head-End Release

the crank end. As the inside lap is less than the outside lap, the exhaust continues longer than the admission.

Fig. 9 shows the relative positions of crank, eccentric, and valve when the exhaust closes on the crank end and compression begins. Between these two positions, the steam is expanding in the head end and exhausting from the crank end. Between the positions of Figs. 9 and 10, both ports are entirely closed, and expansion is taking place in the head end and compression in the crank end. In Fig. 10 is shown the position of the valve for head-end

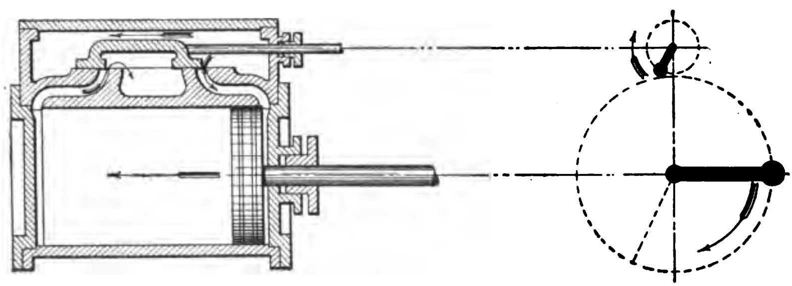


Fig. 11. Position of Valve and Cylinder for Crank-End Admission

release. Fig. 11 shows admission at the crank end of the cylinder and marks the end of crank-end compression.

Effect of Change of Lap. By referring to Figs. 6 to 11, the effect of any change of lap may at once be observed. If the outside lap is increased, the valve must move farther from mid-position before

admission will occur and on the return, after the maximum displacement is reached, the greater outside lap will close the port sooner, and the point of cut-off shown in Fig. 8 will be reached before the crank reaches the angle there shown. A decrease of outside lap will make cut-off later and admission earlier.

On the other hand, if the inside lap is increased, the valve must move farther before release occurs and the crank angle will be greater than that shown in Fig. 10, while on the return to the dotted position, the port will close earlier and make an earlier compression. The crank angle will be less than is there shown. Decreasing the inside lap will cause earlier release and later compression.

Thus we see that it is the outside lap that influences admission and cut-off, and the inside lap that controls release and compression. For this reason the outside lap is often called the *steam lap* and the inside lap is called the *exhaust lap*.

Lead. If a valve having lap is in mid-position, the port is closed and the engine can not start, because no steam can enter the cylinder. That the steam may be ready to enter the cylinder at the beginning of the stroke, it is necessary that the eccentric be set more than 90 degrees ahead of the crank as already mentioned, thus making the eccentric radius take an angular advance EOC, as shown in Fig. 6. In order that the ports and all clearance space may be properly filled with steam at the beginning of the stroke, it is necessary that the valve be displaced from its mid-position an amount slightly greater than the outside lap. With the piston at the end of the stroke, the valve will have a position as shown in Fig. 12, the port being open the distance AB, the lead of the valve. This causes the eccentric to be moved forward a slight amount in excess of the lap angle. This excess is called the *angle of lead.*

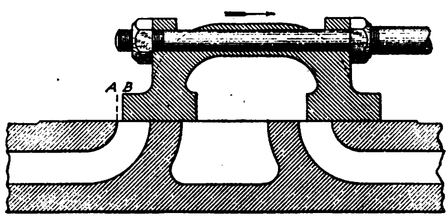


Fig. 12. Position of Valve Showing Lead

In Fig. 13, O_2R_2 represents the position of the crank at the beginning of the stroke, LO_1A_1 the lap angle, and $A_1O_1A_2$ the angle of lead. The eccentric, to give lead, must be set at the angle $R_1O_1A_2$ ahead of the crank or 90 degrees plus the angular advance. In large

high-speed engines, a liberal lead is essential in order that the ports and clearance space may be well filled with steam before the stroke begins. If there is no lead, a portion of the steam will be used in filling these places and full steam pressure will not reach the piston until it is well advanced on the stroke. This will give a sloping admission line, as shown in Fig. 14. Too much lead, on the other hand, will cause too early an admission, as shown in Fig. 15.

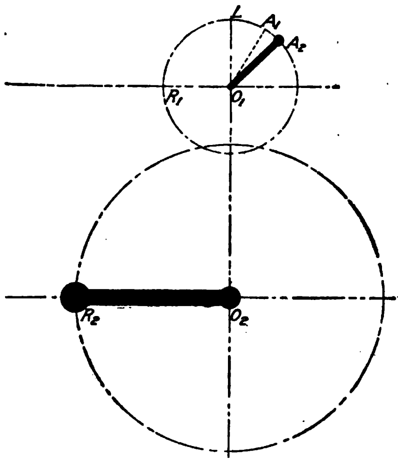


Fig. 13. Diagram Showing Lap Angle and Angle of Lead

If the angular advance is increased, the eccentric will be moved further ahead of the crank, and consequently it will arrive at each of the events sooner than before. If, then, the angular advance is increased, all of the events of the stroke will occur earlier.

Effect of Lead. From the foregoing discussion of lead, it is evident that its effect is to permit steam to enter the cylinder before the end of the stroke, which tends to provide an abundance of steam behind the piston when starting the return stroke and throughout the period of admission. It also promotes smooth running of the engine by furnishing a cushion or retarding force to the moving parts, thereby eliminating the "knocks" or "pounds" incident to lost motion. Since the effect of lost motion depends upon the weight

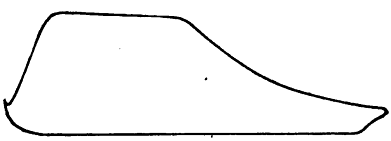


Fig. 14. Indicator Diagram Showing Effect of No Lead

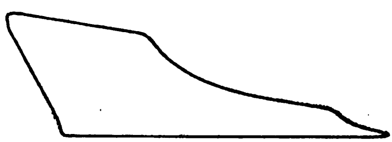


Fig. 15. Indicator Diagram Showing Too Early Admission

and velocity of the reciprocating parts, it is evident that the amount of lead required will vary for different engines and for the same engine running at different speeds. The exact amount of lead can not be determined except by trial and by use of the steam engine indicator.

When experimenting for the determination of the proper amount of lead for a specific case, it will be necessary to gradually increase the angular advance until smooth running is obtained. After this result is obtained, indicator cards should be taken to see if the lead is excessive, in which case the valve must be adjusted until the desired conditions are obtained. Since lead permits steam to act against the piston before the end of the stroke, it results in negative work, hence the amount of lead should not be excessive. An amount of lead sufficient to insure the filling of the clearance space is permissible, but very much more than this is detrimental to the economic performance of the engine.

Analytical Summary of Valve Terms. Thus far in discussing the plain slide valve, a number of terms have been used that are of primary importance and must be thoroughly understood in order to properly grasp much that is yet to be studied. It seems advisable, therefore, that a recapitulation of the terms used be presented.

Mid-Position. A valve is said to be in mid-position when the center of the valve and valve seat coincide. When in this position, the steam ports are all closed.

Displacement. The displacement or a valve is the amount the valve has been moved either to the right or left of its mid-position. In Fig. 4, the valve has moved to the right a distance equal to the width of the steam port, hence in this instance the displacement of the valve is equal to the width of the steam port.

Valve Travel. The travel of the valve is the distance the valve travels in moving from one extreme position to the other. The travel of the valve is twice the eccentricity, or throw of the eccentric.

Eccentricity. The eccentricity, or throw of the eccentric, is the distance between the center of the shaft and the center of the eccentric. It is equivalent to a crank, the length of which is one-half the valve travel. For instance, if the valve travel of an engine is 6 inches, the eccentricity, or throw of the eccentric, would be 3 inches, or one-half of the valve travel.

Lap. The amount that the valve extends over the steam port when in mid-position is called steam lap or often spoken of as the lap of the valve. The steam lap is equal to $O\,C$ in Fig. 5. In Fig. 5, it is obvious that when the valve is in mid-position, the distance $D\,I$ is called exhaust lap. Steam lap and exhaust lap are frequently

spoken of as outside and inside lap, respectively. The effect of the exhaust lap is to delay exhaust and hasten compression.

Very frequently a valve does not have any exhaust lap and there is a small port opening between the cylinder and the exhaust cavity when the valve is in mid-position, as shown at A, Fig. 19. In such a case, the valve is said to have *inside clearance.* The effect of inside clearance is opposite to that of exhaust lap, namely, it delays compression and hastens exhaust, and insures a minimum amount of back pressure.

Lead. By the term *lead* is meant the amount the steam port is open when the engine is on either dead center.

Angle of Advance. It was noted in Fig. 1 that the crank and eccentric were exactly 90 degrees apart and that admission occurred at the beginning and cut-off at the end of the stroke. On account of economic reasons, this is not a good arrangement. Hence we find that the valves on all engines have lap and are set to give the necessary amount of lead. In order to obtain lead when the engine is on dead center with a valve having lap, it is necessary to turn the eccentric ahead, in the direction the engine is to run, through such an angle that the valve will be displaced by an amount equal to the lap plus the lead. The angle measuring this displacement is the sum of the angle of lap and the angle of lead. If there is no lead, this angle would be decreased by the angle of lead. The sum of the angle of lap and the angle of lead is frequently designated as the angle of advance. The angularity between the eccentric and the crank then becomes equal to 90 degrees, plus or minus the angle of advance according to the type of valve and gear.

Inequality of Steam Distribution. In the valve diagrams thus far considered, the events of the stroke have been discussed for each end separately, without reference to the relation of similar events on the other side of the piston. If the connecting rod were of infinite length, so that it would always remain parallel to the center line of the engine, the distribution would be the same for both ends of the cylinder. In practice, the connecting rod varies from four to eight times the length of the crank, which causes the connecting rod always to be at an angle to the center line of the engine when the engine is off dead center, and for a given crank angle makes the piston displacement greater at the head end than at the crank end.

To Find Displacement of Valve. The circle, Fig. 16, represents the path of the eccentric center during a complete revolution of the engine. OC represents the crank, and OR the corresponding position of the eccentric. The diameter XY represents the extent of the valve travel. Since the eccentric rod is so long in comparison to the eccentricity, we make no appreciable error by assuming it always to be parallel to the center line of the engine. When the eccentric is at OL, the valve is in mid-position. At OR the valve has moved from mid-position an amount ON, found by dropping a perpendicular from R to the center line XY. If the angularity of the connecting rod could be neglected, the piston displacement could be found in the same manner.

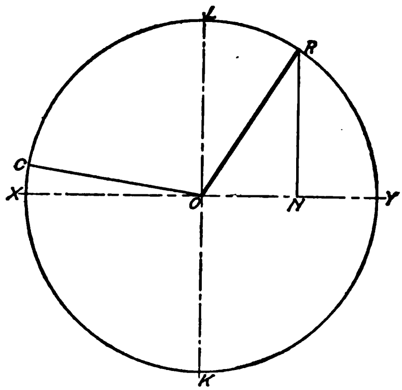


Fig. 16. Eccentric Circle Showing Relative Positions of Crank and Eccentric

To Find Displacement of Piston. To find the displacement of the piston, a diagram as shown in Fig. 17 must be drawn. In this figure, AB represents the cylinder, P_1 the piston, H_1 the crosshead, H_1R the connecting rod, and OR the crank. Suppose now the engine should stop and the piston be clamped in this position. The piston displacement would be represented by AP_1. If the crank pin at R should now be loosened so as to allow the connecting rod to fall to a

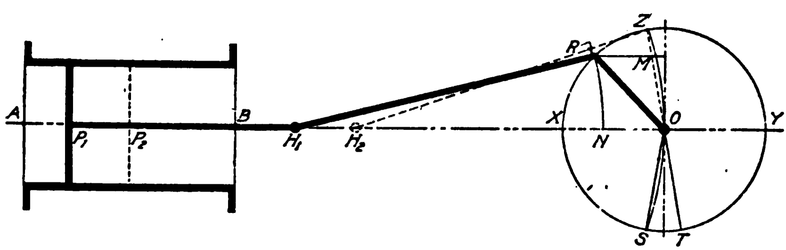


Fig. 17. Diagram for Finding Displacement of Piston

horizontal position, the point R would describe the arc of a circle RN, and XN would represent the piston displacement and would be equal to AP_1. Suppose now that in this disconnected way, the

piston, crosshead, and connecting rod were moved forward until the end of the rod came to O. P_1 would then be at P_2 and the piston would be in the middle of its stroke. Now suppose the end of the rod were swung up to its proper position on the crank-pin circle, the piston remaining stationary. The end of the rod would describe an arc OZ; the crank pin would be at Z, less than a quarter revolution from X; while the piston would be in the middle of its stroke.

Suppose this engine were running with cut-off at half stroke on the head end, and that XOZ represented the corresponding crank angle. On the return stroke, the valve would cut off at the same crank angle YOT, which is equal to XOZ, and OT would represent the crank position for cut-off on the return, or crank-end, stroke. The piston, as we have just seen, will not be at half stroke except

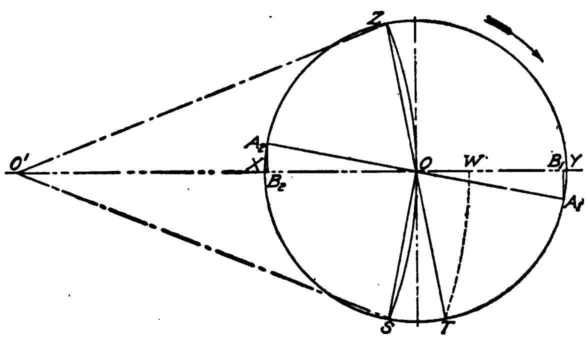


Fig. 18. Crank and Eccentric Diagram for Engine Shown in Fig. 17

when the crank is at OZ or OS. Consequently, the crank position OT is less than half stroke and cut-off occurs earlier at crank end than at head end. When the crank is at OZ, the eccentric will be at OA_1, Fig. 18, assuming the valve to have no lap, and the valve displacement will be OB_1. When the crank is at OT, the eccentric will be at OA_2 and the valve displacement will be OB_2, which is equal to OB_1, the displacement of the valve at cut-off on the head end. The piston displacement will be OX on the head end and WY on the crank end when cut-off occurs. If the connecting rod always remained parallel to the center line, the cut-off would be the same at both ends.

Compensation of Cut-Off. It has been pointed out that lengthening the outside lap makes the cut-off earlier, and that shortening

the lap makes it later. The cut-off in the case just cited may then be equalized by altering the outside laps. If we increase the outside lap on the head end, or decrease the crank-end lap, the inequality will be less. By changing either or both of the laps the proper amount, the cut-off may be exactly equalized.

But altering the outside lap changes the lead, as has already been explained. If the lap is increased on the head end, the lead will be less than on the crank end. If the lead becomes too small on the head end, the angular advance may be increased but the inequality of lead will still remain, for this increase of angular advance will increase the lead at the crank end as well as at the head end, and by hastening all the events of the stroke may give a bad steam distribution if the proper care is not taken.

Unequal lead is of less consequence on a low-speed engine than on a high-speed engine. On low-speed engines, the cut-off may be

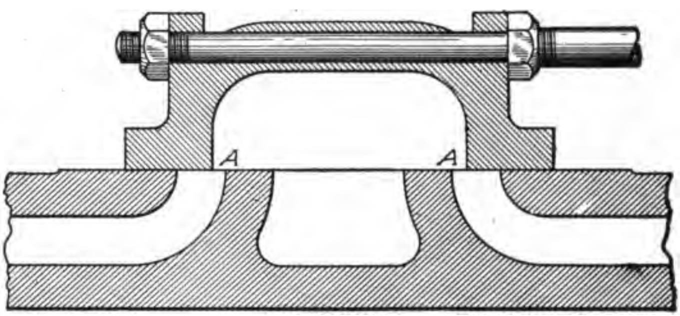


Fig. 19. Valve in Mid-Position Showing Inside Clearence, or Negative Lap

equalized at the expense of lead with beneficial results, but on high-speed engines, this is not true. A high-speed engine requires more lead than a low-speed engine, for there is relatively less time in each stroke for the clearance space to fill with steam.

If both inside laps are equal, compression will not occur equally at both ends. To equalize it, the inside laps may be changed in the same manner as the outside laps are changed to equalize the cut-off. By altering these inside laps to equalize compression, it may happen that the lap is reduced enough to leave the exhaust port open when the valve is in mid-position. The amount of this opening is called *inside clearance*, or *negative lap*. This is illustrated at *A*, Fig. 19.

Rocker. Sometimes it happens that the valve stem and eccentric rod can not be so placed that they will be in the same straight line; or it may be that the travel of the valve must be so great as to require an excessively large eccentric. In such cases, a rocker may be used.

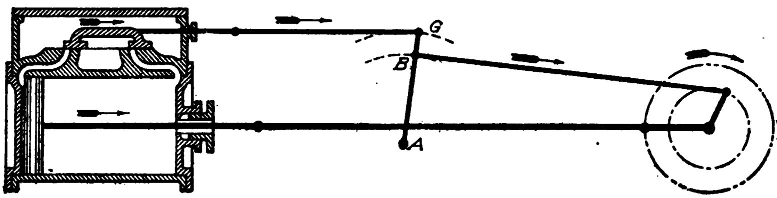


Fig. 20. Valve with Rocker Arrangement

Fig. 20 shows a valve that is not in line with the eccentric. An instance where this occurs is in horizontal engines when the valve is set on top of the cylinder instead of on one side. By means of the rocker *A G*, the valve may receive its proper motion.

In case it is more convenient to place the pivot of the rocker arm between the connections to the valve stem and those of the eccentric rod, such an arrangement as is shown in Fig. 21 may be used. Here it will be noticed that the valve stem and eccentric rod are moving in opposite directions and in order to give the valve the same motion as in Fig. 20, the eccentric must be moved 180 degrees ahead of the position there shown.

If *A B* is less than *A G*, the valve travel will be greater than twice the eccentricity, in proportion as *A G* is greater than *A B*. In all

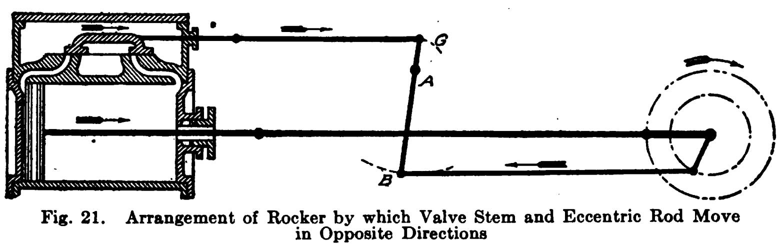


Fig. 21. Arrangement of Rocker by which Valve Stem and Eccentric Rod Move in Opposite Directions

cases, the valve travel is in the same proportion to twice the eccentricity as *A G* is to *A B*. Thus, if the valve travel is $4\frac{1}{2}$ inches, *A B* is 15 inches, and *A G* is 18 inches, then $\frac{15}{18} \times 4\frac{1}{2} = 3\frac{3}{4}$ inches, will equal twice the eccentricity.

A valve gear may be so laid out as to make both the cut-off and the lead equal for both ends of the cylinder. This may be done by

a proper proportion between the rocker arms, and a careful location of the pivot of the rocker. The eccentric must then be set accordingly. In this manner, the Straight Line engine equalizes the cut-off and lead. A discussion of this method will be considered later.

VALVE DIAGRAMS

Zeuner Diagrams. In order to study the movements of the valves, the effects of lap, lead, eccentricity, etc., diagrams of various sorts have been devised. By the use of diagrams we may acquire

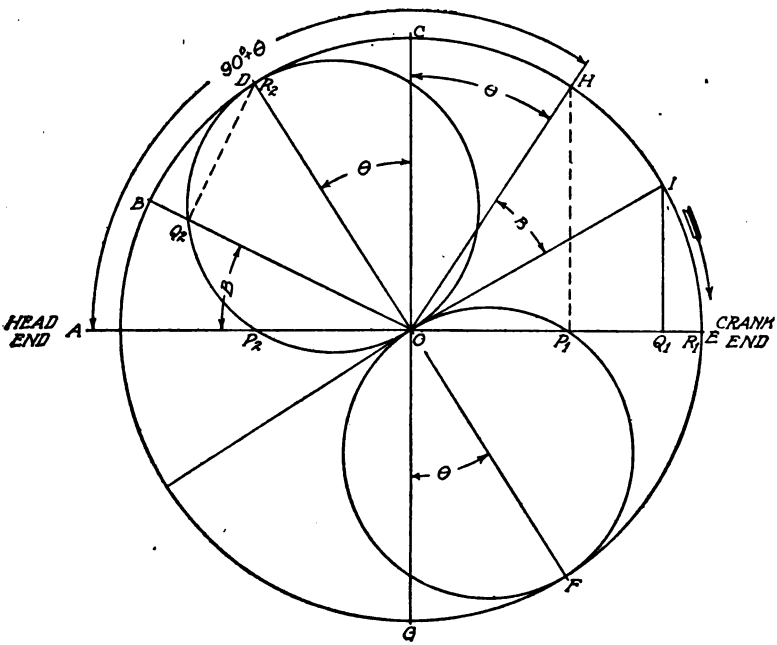


Fig. 22. Zeuner Diagram for Valve Analysis

a knowledge of the valve motion without the complex mathematical expressions that such a discussion would entail. The most useful of these various diagrams is that devised by Zeuner and, to avoid complexity, we shall confine ourselves to a discussion of this diagram alone. The eccentric rod is assumed to be of infinite length, and the positions of the crank are shown on the diagrams. The displacement of the piston can easily be found if the ratio of crank to connecting rod is known.

The function of the Zeuner diagram is to show the relation between the valve positions and crank positions. This relation being known, it is a simple matter to obtain the eccentric and piston positions.

In Fig. 22, $A\,O\,E$ represents the valve travel, and the center of the eccentric will move in the circle $A\,C\,E\,G$. It is assumed, also, that this circle represents the path of the crank center, hence this circle will be known as the *crank and valve circle*. OA is the position of the crank and $O\,H$ is the corresponding position of the eccentric, when the engine is on the head-end dead center. Since this valve has lap, and since admission must occur before the end of the stroke, it is evident that the eccentric must precede the crank by 90 degrees plus the angle of advance θ. From H drop a perpendicular line upon the center line $A\,O\,E$, thus locating the point P_1. The distance OP_1 is the amount the valve has been moved to the right of its mid-position when the crank is on dead center. Since the diagram gives the relation between crank and valve positions, the displacement of the valve $O\,P_1$ can be laid off from O on the crank position OA, thus establishing the point P_2. Turn the crank through an angle B to the position $O\,B$. The eccentric will move through the same angle and will be found at I. Draw the perpendicular line IQ_1, and OQ_1 represents the displacement of the valve for the crank position $O\,B$. Lay off OQ_1 on $O\,B$, establishing the point Q_2. Continue the rotation of the crank until the point D is reached. The eccentric then will be found at E, and the valve will have its greatest displacement $O\,R_1$ to the right of its mid-position. It is evident that $O\,R_1$ is equal to $O\,D$. If the rotation of the crank be continued in the direction of the arrow, the valve will return from its extreme position on the right and will approach its mid-position. By locating on the various crank positions the corresponding valve displacement, a series of points as P_2, Q_2, R_2, etc., will be obtained, all of which will lie on the circumference of a circle, as $OP_2Q_2R_2$, the diameter $O\,D$ of which will make an angle θ equal to the angle of advance laid off to the left of the vertical $O\,C$. If this operaation be continued for a complete revolution, a series of points will be established in the lower quadrant, establishing a circle OP_1F, the diameter of which will be a continuation of OD and, therefore, will make an angle θ with the vertical but will lie on the right of

the vertical line COG. These two circles are called *valve circles*, and they represent the movement of the valve to the right and left of its mid-position and, as previously stated, represent the amount the valve has moved for any crank position such as OB.

Having established the valve circles, it is a simple matter to obtain the valve displacement for the position OB, which, in this case, would be the distance OQ_2 cut off from OB by the valve circle. It can be proven that OQ_2 is the valve displacement by comparing the two right triangles OIQ_1 and ODQ_2. They are equal because

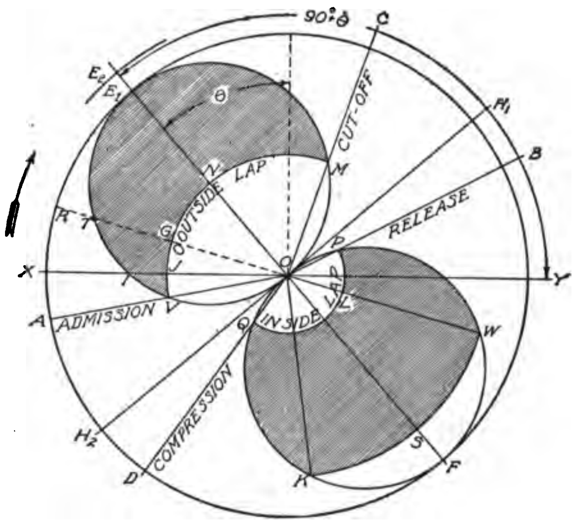


Fig. 23. Diagram Showing Study of Valve Motion for Head End Only

they are similar and have two corresponding sides OD and OI equal. Therefore, OQ_2 equals OQ_1. This being true for any crank position, it is true for all crank positions.

Study of Valve Motion from Diagram. Now that the truth of our proposition has been proved, let us see how we may study the valve motion from such a diagram. In Fig. 23 let XY represent the valve travel; then the circle XE_1YF will represent the path of the center of the eccentric. Let θ be the angle of advance and lay off E_1O toward the crank, making an angle θ with the vertical. Produce E_1O to F, and on OE_1 and OF as diameters draw the valve circles as shown. With O as a center and OV, equal the outside lap, as a

radius, draw an arc intersecting the upper valve circle at V and M. Lay off $O\,P$ equal to the inside lap and with O as a center and $O\,P$ as a radius, draw an arc intersecting the valve circle at P and Q. Draw the crank-position line $A\,O$ passing through V. Then, when the crank is in this position, the displacement of the valve is equal to $O\,V$ (the outside lap) and the steam is ready to enter the cylinder. This is the position of the crank at admission, and the crank angle XOA is called the *lead angle*. The valve has lead and, therefore, the admission takes place before the end of the stroke. When the crank reaches the position OE_1, the displacement of the valve is equal to the eccentricity OE_1, and is at a maximum. Further motion of the piston causes the valve to move toward mid-position until, at the crank position $O\,C$, the displacement $O\,M$ is again equal to the outside lap and the valve has reached the point of cut-off. When the position $O\,H_1$ is reached, the crank line is tangent to both valve circles and there is no displacement of the valve. At this point, the valve is in mid-position.

Further crank movement draws the inside lap toward the edge of the exhaust port until, at the crank position $O\,B$, the displacement is equal to OP (the inside lap) and release begins. At OF the maximum valve displacement is again reached and the valve moves in the opposite direction until at $O\,D$ its displacement from mid-position is again equal to $O\,Q$, equals OP the inside lap, and compression takes place. At $O\,H_2$ the valve is again in mid-position. At OX the displacement of the valve is OI, but since the valve has to move the distance OJ before the port begins to open, IJ must represent the port opening when the crank is on dead center, and by definition we know that lead is the amount of port opening at this position. Therefore, $I\,J$ represents the lead.

At the position R, the port is open an amount equal to TG; at E_1 the opening is a maximum equal to E_1N; at C the port is on the point of closing and there is no opening. If LW represents the total width of the steam port, the exhaust port will be open wide when the displacement of the valve is equal to $O\,W$ and it will remain wide open while the crank swings from $O\,W$ to $O\,K$.

If the width of steam port in addition to the outside lap were laid off on the other valve circle, it would fall at E_2. For the admission port to be wide open, the displacement of the valve would have

to be equal to OE_2 which is more than the maximum displacement. This shows that in this case the steam port is never fully open and that the left-hand edge of the valve overlaps the right-hand edge of the port by an amount equal to E_1E_2 when the valve has reached its maximum displacement.

Fig. 23, with its two valve circles, shows the diagram for the head end of the cylinder only. The crank-end diagram would be similar except that the laps might not be equal to those of the head end.

Properties of Zeuner Diagrams. The Zeuner diagram deals with admission, cut-off, release, compression, lead, valve travel, angle of

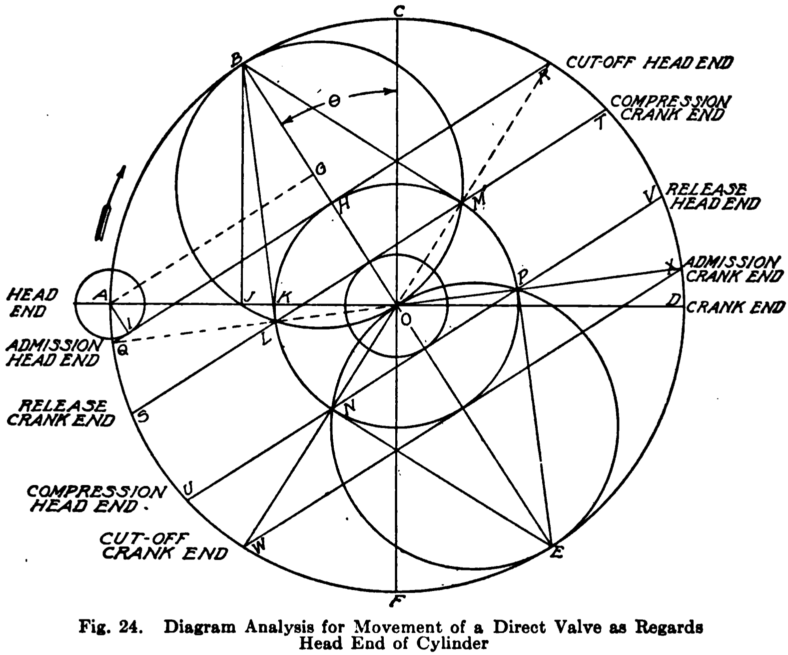


Fig. 24. Diagram Analysis for Movement of a Direct Valve as Regards Head End of Cylinder

advance, maximum and minimum port opening, steam lap, and exhaust lap. Generally, if four of these be given, the others can be found. It is evident, therefore, that there are a great many possible combinations, hence it is necessary to have definitely in mind and clearly understood the properties of the Zeuner diagram. The proofs given are for the movement of a direct valve as regards the head end of the cylinder. All letters refer to Fig. 24.

(1) The figure is symmetrical on the line BE. In the semicircles OLB and OMB, OL equals OM, each being the radius of the steam-lap circle. Since OL equals OM, the arcs which they subtend are equal, therefore, the arcs LJB and MB are equal. This makes the angles LOB and MOB equal because they are measured by equal arcs. Therefore, BO bisects the angle LOM, and in a similar way it can be proved that OE bisects the angle NOP.

(2) The line BM is perpendicular to OMR and is tangent to the steam-lap circle. The angle BMO is a right angle because it is inscribed in a semicircle. Therefore, BM is tangent to the steam-lap circle and is perpendicular to the crank position OMR.

(3) The line joining the admission and cut-off points for the head end is perpendicular to BO and is tangent to the steam-lap circle.

The triangle QOR is an isosceles triangle and, as demonstrated above, BO bisects the angle QOR, hence BO is perpendicular to the base QR. To prove that QR is tangent to the steam-lap circle, it is necessary to show that the distance OH measured on BO is equal to OM, the radius of the steam-lap circle. The right triangles BOM and HOR are equal, having two sides equal and one common angle. Hence, OH is equal to OM.

(4) The line BJ is perpendicular to AO. The angle BJO is a right angle, being inscribed in a semicircle.

(5) The radius of the circle AI with center at A and tangent to QR, is equal to the lead JK.

From the center A draw AG parallel to IH. In the right triangles BJO and AGO, the angle AGO equals BJO, being right angles. BO equals AO. The angle AOH is common to both triangles, therefore, they are equal. Hence, OJ equals OG. But OK equals OH. Therefore, GH equals JK, equals AI, which is the lead.

By using Fig. 24 at all times as a reference figure and bearing in mind the things it tells, no great difficulty should be encountered in solving problems. To illustrate the principles set forth above and to give an idea of the practical use of the Zeuner diagram, several problems will be worked out as an indication of what may be done.

ILLUSTRATIVE PROBLEMS

In designing a slide valve, a few of these variables depend upon the conditions under which the engine is to run. For instance, the valve travel is limited, cut-off must be at a certain point, and the engine must have a certain lead. Then, with the aid of a Zeuner diagram, the remaining proportions of the valve may be determined.

EXAMPLE 1. Given a valve travel of 3 inches, exhaust lap of $\frac{3}{4}$ inch, angular advance of 35 degrees, and crank angle at cut-off of 115 degrees. Determine the laps, the lead, and the crank angles at admission, compression, and release.

SOLUTION. In Fig. 25, let $X\ Y$ represent the valve travel of 3 inches. Draw $O\ M$ perpendicular to $X\ Y$, and on $X\ Y$ as a diameter draw the circle

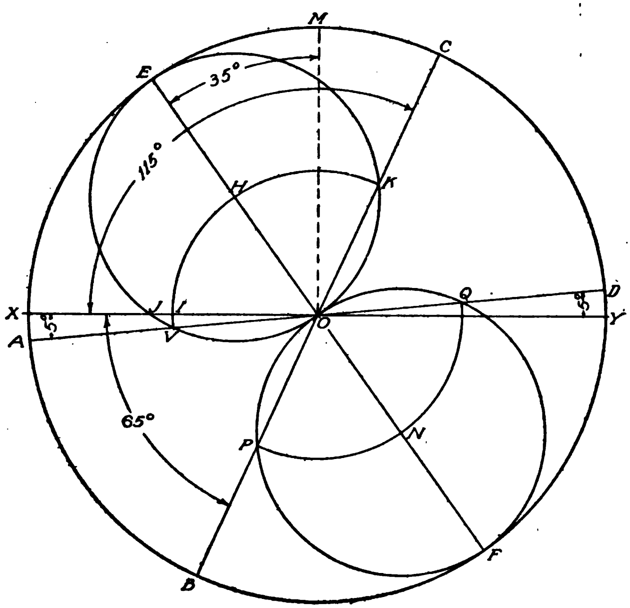


Fig. 25. Zeuner Diagram for Finding Laps, Lead, and Crank Angles

$X\ M\ Y\ F$ representing the path of the center of the eccentric as it revolves about the shaft. Lay off the angle $M\ O\ E$ to represent the angular advance of 35 degrees so that the angle $X\ O\ E$ is equal to 90 degrees minus the angular advance. Produce $E\ O$ to F. Then on $O\ E$ and $O\ F$ as diameters, draw the valve circles. The eccentricity $O\ E$ or $O\ F$, if no rocker is used, will be half the valve travel. Lay off the angle $X\ O\ C$ to represent the crank angle at cut-off of 115 degrees, and $O\ K$ will then represent the distance of the valve from mid-position when cut-off takes place. This distance we know is the outside lap. Draw the arc $K\ I$, known as the lap circle, and it will cut the valve circle again at V. When the valve is again the distance $O\ V$, the out-

side lap from mid-position, admission will take place. Draw the line $O\,V\,A$ and this will represent the position of the crank at admission.

When the crank is at $O\,X$, the valve displacement is equal to $O\,J$. This is at dead center, and the valve is open the amount $I\,J$, for it has moved this distance more than the outside lap. Therefore, $I\,J$ is the lead for this end.

Now on the other valve circle, draw the arc $P\,Q$ with the inside lap ($\frac{3}{4}$ inch) as a radius. It will cut the valve circle at P and Q. When the valve displacement is equal to $O\,Q$, the exhaust port has just commenced to open, and the engine is at release. In the same way, when the valve displacement is equal to $O\,P$, the port begins to close and the engine is at compression. $O\,Q\,D$ represents the crank position at release and $O\,P\,B$ the crank position at compression.

The results are tabulated as follows:

Outside lap $O\,K$	$=\frac{3}{4}$ inch
Angle of lead $X\,O\,A$	$=5$ degrees
Linear lead $I\,J$	$=\frac{3}{32}$ inch
Max. port opening for admission $H\,E$	$=\frac{3}{4}$ inch
Crank angle at release $X\,O\,D$	$=185$ degrees
Crank angle at compression $X\,O\,B$	$=$ 65 degrees
Max. port opening for exhaust $F\,N$	$=\frac{3}{4}$ inch

Fig. 25 is drawn full size, and all of the above measurements may readily be verified. This figure is drawn for the head end only. If the crank angle at cut-off is the same on both ends, the Zeuner diagram for the crank end will be exactly like Fig. 25.

EXAMPLE 2. Given a lead $\frac{1}{16}$ inch, valve travel 3 inches, steam lap (h. e. and c. e.) $\frac{7}{8}$ inch, exhaust lap (h. e. and c. e.) $\frac{3}{16}$ inch. Let $\frac{R}{L}$, that is, the ratio of the length of the crank to the connecting rod, equal $\frac{1}{6}$. Construct the Zeuner diagram and find all the events for both the head and crank ends in per cents.

SOLUTION. Construct the valve travel circle $A\,C\,D\,F$, Fig. 26, with a radius of $1\frac{1}{2}$ inches; the steam-lap circle with a radius $O\,H$ of $\frac{7}{8}$ inch; and the exhaust lap circle with a radius $O\,R$ of $\frac{3}{16}$ inch. The steam-lap circle cuts the crank position for h. e. dead center at the point K. From K lay off the distance $J\,K$ to represent the lead of $\frac{1}{16}$ inch. At A, construct the lead circle with a radius of $\frac{1}{16}$ inch. From the properties of the Zeuner, we know that where a perpendicular erected at the lead point J cuts the valve travel circle as at B, the line $B\,O$ is the diameter of the valve circle and the angle $C\,O\,B$ is the required angle of advance. We also know that a line drawn perpendicular to $B\,O$ and tangent to the steam-lap circle cuts the valve travel circle at the points of admission and cut-off, respectively. Therefore, draw $S\,T_2$ so it will be tangent to the steam-lap circle and perpendicular to $B\,O$ at H. The points S and T_2 are the points of head-end admission and cut-off, respectively. It is to be noted, also, that this line $S\,T_2$ is tangent to the lead circle, which fulfills another condition of the property of the Zeuner.

To locate the other events for the head and crank ends, draw lines perpendicular to $B\,O\,E$ and tangent to the steam- and exhaust-lap circles, and the points where these lines cut the valve travel circles will be the required

points. In the same manner, the several other points in the figure have been located.

To find the per cent of stroke at which the several events occur, take a radius proportionately equal to the length of the connecting rod and describe the arcs shown. As $\frac{R}{L}$ is $\frac{1}{5}$, L equals 5 R. But R is one-half the valve travel, *i.e.*, 1.5 inches.

$$\therefore \quad L = 5 \times 1.5$$
$$= 7.5 \text{ inches}$$

Now, with a radius of 7.5 inches and with a center on the horizontal line through the center of the valve travel circle produced to the left of the

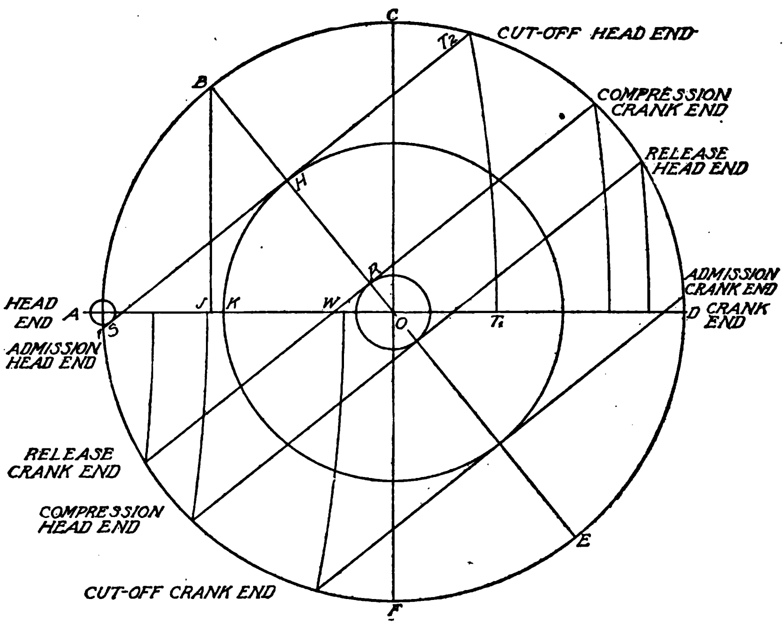


Fig. 26. Diagram for Finding Events for Head and Crank Ends; Lead, Valve Travel, and Laps being Given

vertical line $C\,F$, sweep the arcs shown from the points of admission, cut-off, etc., on the head and crank ends. Remembering that the head-end events are measured from the head-end dead center and the crank-end events from the crank-end dead center, measure the distance $A\,T_1$. This distance, 2.03 inches, divided by the valve travel 3 inches, and multiplied by 100, gives the per cent cut-off on the head end, that is,

$$\frac{2.03}{3} \times 100 = 67\% \underset{\text{h.e.}}{\text{cut-off}}$$

In like manner, measure the distance $D W$ for the crank-end cut-off, which we find is 1.8 inches. Then

$$\frac{1.8}{3}\times 100 = 60\% \text{ cut-off}_{\text{h.e.}}$$

Continuing this procedure for the other events, the final results obtained from the diagram will be

EVENT	HEAD END	CRANK END
Admission	98 per cent	98 per cent
Cut-off	67 per cent	60 per cent
Release	93 per cent	91 per cent
Compression	16 per cent	$12\frac{1}{2}$ per cent

Angle of advance $\theta = 40$ degrees

EXAMPLE 3. Given an engine having 30 per cent cut-off on the head end; maximum port opening of $\frac{3}{8}$ inch; and lead on the head end $\frac{1}{16}$ inch. The laps are to be equal; compression on the head end is 25 per cent; and $\frac{R}{L}$ equals $\frac{1}{4}$. Construct the Zeuner diagram.

SOLUTION. In Fig. 27, lay off $E F$ to represent the maximum port opening $\frac{3}{8}$ inch; $F G$ the lead $\frac{1}{16}$ inch; and erect perpendiculars $E J$, $G H$, $F I$. On any point as O on the line GH, draw a trial circle such as $A B C D$, which in this case was assumed to be 1 inch in diameter. Since cut-off on the head end occurs at 30 per cent of the stroke, locate the direction of the crank position $O P$ for this position. This direction will hold for any valve travel. Draw $O M$ perpendicular to $O P$, cutting $F I$ at K. Bisect the angle $F K M$ by $K N$. On $K N$ as a center line, find by trial a radius and center, such that a circle when described will pass through O and be tangent to $E J$. The center is found to be at L and the distance $O L$ is the radius of the required valve circle. With L as a center, draw a circle tangent to $F I$ and $K M$. Such a circle will be the required steam-lap circle. To demonstrate why this construction is correct, it is only necessary to refer to the properties of the Zeuner diagram as given in connection with Fig. 24. Here it is shown that a line drawn perpendicular to the crank position for the point of cut-off and tangent to the steam-lap circle cuts the valve travel circle at the extremity of the valve circle, as at B. Hence, $O M$ fulfills this condition, which gives the extremity of the required valve travel circle at O. In Fig. 24, it is also evident that the steam-lap circle is tangent to the perpendicular to the crank position for the given cut-off and is also tangent to a perpendicular to the horizontal center line drawn at the extremity of the maximum port opening. Therefore, this condition was fulfilled in establishing the required lap in Fig. 27. Having obtained the valve travel and lap, it remains to complete the diagram in order to determine the other conditions. In Fig. 28, the circles $A B C D$ and $a b c d$ are constructed on a diameter of $4\frac{11}{16}$ inches and $4\frac{1}{16}$ inches, respectively, the former being the value of the valve travel and the latter twice the steam lap, as found in Fig. 27. Locate the head-end cut-off at 30 per cent and draw the lead circle with a radius of $\frac{1}{16}$ inch. Locate the head-end compression of 25 per cent at I. Draw $G H$ tangent to the

steam-lap and lead circles cutting the valve travel circle at G, thus establishing the head-end admission. From the properties of the Zeuner diagram as discussed on pages 21 and 22, we know the diameter of the valve circles will be on a line bisecting the angle GOH. Draw the line $F\,O\,E$ bisecting this angle; this line will be perpendicular to $G\,H$. Having established $F\,O\,E$ and bear-

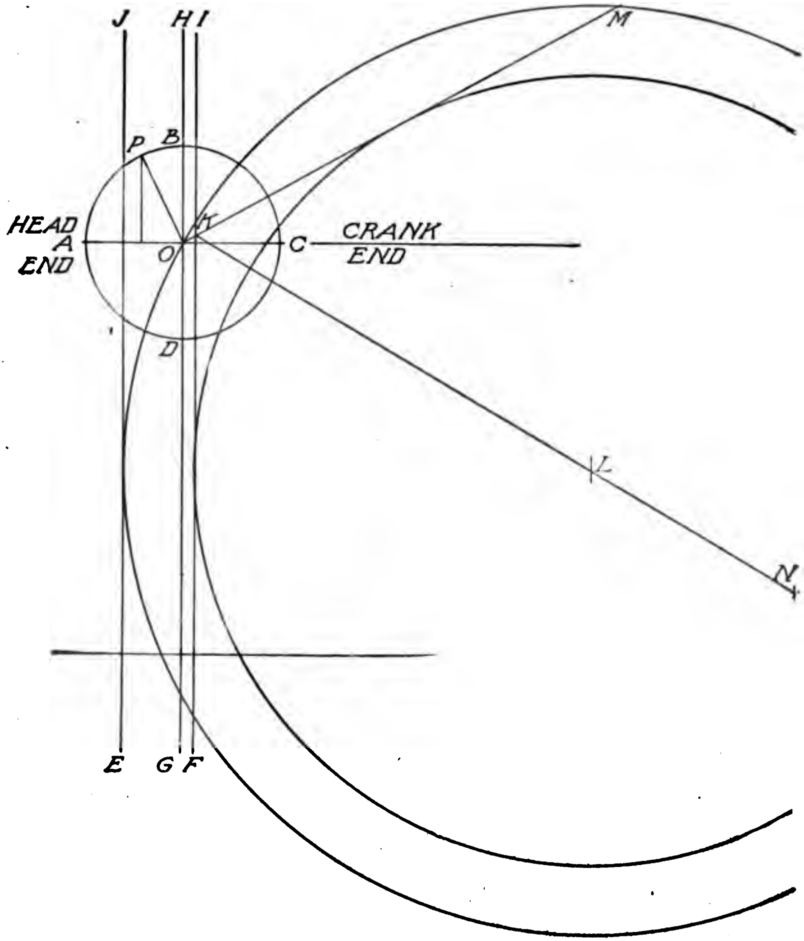


Fig. 27. Diagram for Engine with Thirty Per Cent Cut-Off, Laps Equal, Compression Twenty-Five Per Cent

ing in mind the demonstrations previously given, it is evident that a line drawn from the point of compression on the head end I perpendicular to $F\,O\,E$ will cut the valve travel circle at J, the point of head-end release. It is to be noted, however, that the line joining the points of release and compression on the head end lies on the same side of the center O as does the line joining the points

of admission and cut-off for the head end. This relation being opposite to that found in Fig. 24 means that instead of having exhaust lap with this valve, there is inside clearance equal to $O\,N$. With O as a center and $O\,N$ as a radius, describe the clearance circle and complete the Zeuner by drawing the parallel lines $K\,L$ and $P\,Q$, thus locating the remaining events of the stroke. In order to obtain the per cents of the events of the stroke, proceed as in Example 2.

The results are tabulated as follows:

Steam lap	$=2\frac{1}{32}$ inches
Inside clearance	$=\frac{5}{16}$ inch
Valve travel	$=4\frac{13}{16}$ inches
Angle of advance	$=62$ degrees
Admission on both h.e. and c.e.	$=99$ per cent (approx.)
Cut-off c.e.	$=20$ per cent
Compression c.e.	$=18$ per cent
Release c.e.	$=60$ per cent
Release h.e.	$=72$ per cent

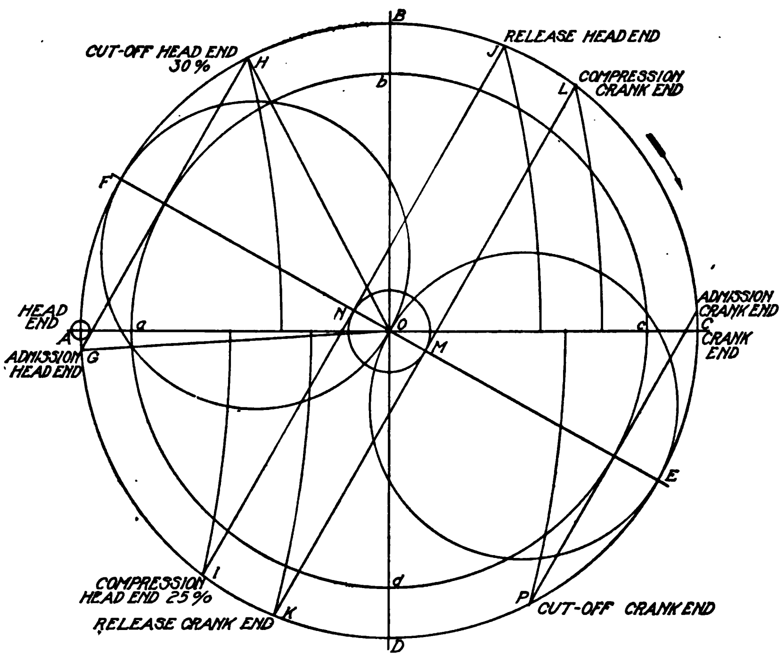


Fig. 28. Diagram for Example 3 to Determine Admission, Compression, and Release at Crank and Head Ends

The preceding problems involve nearly all of the properties of the Zeuner diagram and, if completely mastered by the student, should make the solution of other problems very much easier.

Effect of Changing Lap, Travel, or Angular Advance. We are now in a position to consider more in detail the effect of changing

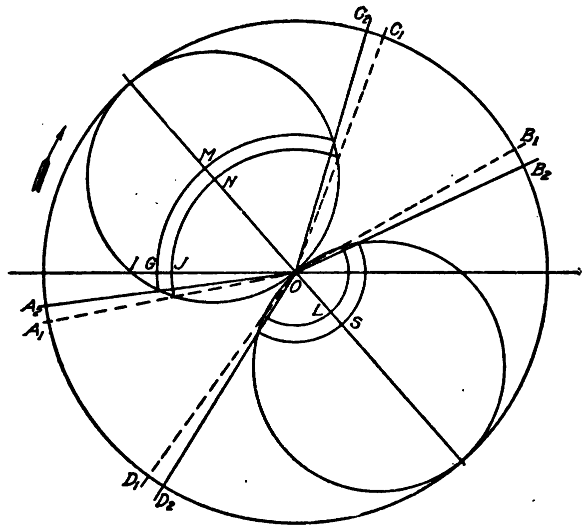


Fig. 29. Study of Effect of Changing Valve or its Setting

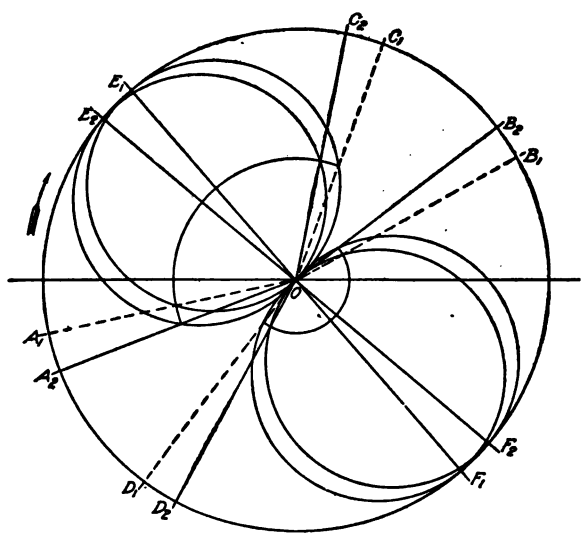


Fig. 30. Study of Effect of Changing Angle of Advance

in any way either the valve or the setting. Let us consider Fig. 29, which is in every way like Fig. 23 except that all unnecessary

TABLE I

Effect of Changing Lap, Travel, and Angular Advance

Event	Increasing Outside Lap	Increasing Inside Lap	Increasing Travel	Increasing Angular Advance
Admission	Is later Ceases sooner	Not changed	Begins earlier Continues longer	Begins earlier Same period
Expansion	Is earlier Continues longer	Beginning unchanged Continues longer	Begins later Ceases sooner	Begins earlier Same period
Exhaust	Unchanged	Occurs later Ceases sooner	Begins earlier Ceases later	Begins earlier Same period
Compression	Begins at same point Continues longer	Begins sooner Continues longer	Begins later Ceases sooner	Begins earlier Same period

letters and lines are omitted to avoid confusion. If the outside lap, or steam lap, is *increased* an amount equal to $N M$, the admission will take place later, viz, at crank position $O A_2$; the lead will be reduced to $I G$ and cut-off will take place earlier, viz, at $O C_2$.

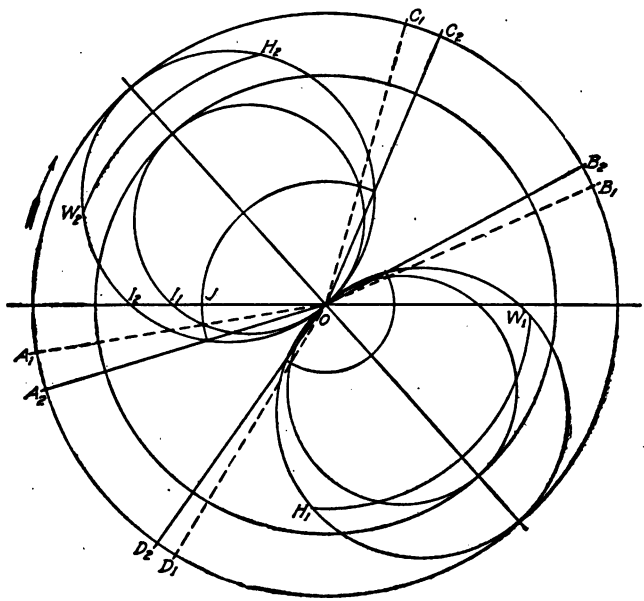


Fig. 31. Study of Effect of Changing Eccentricity

If the outside, or steam lap, is *reduced* a like amount, the contrary effects will be observed. If the inside lap, or exhaust lap, is increased an amount equal to $L S$, the release will take place later at the crank position $O B_2$, and compression will take place earlier at $O D_2$. The

contrary effect will be observed by decreasing the inside lap, or exhaust lap.

If the angular advance is increased, all the events will occur earlier, as is evident from Fig. 30. The crank revolves in the direction indicated by the arrow and $O\,A_2$ (new position of admission) is *ahead* of $O\,A_1$ the old position.

If the eccentricity is increased, Fig. 31, the valve travel will increase and admission will take place earlier at $O\,A_2$; the lead will be increased an amount equal to I_1I_2, and cut-off will take place later at $O\,C_2$. Release will be earlier at OB_2 and compression will be later at $O\,D_2$. The upper valve circle will now cut the arc drawn from O as a center, with a radius equal to the outside lap plus the width of steam port, in the points W_2 and H_2, and the admission port will be open wide while the crank is moving from $O\,W_2$ to $O\,H_2$. Similarly, the lower valve circle cuts the arc drawn from O as a center, with a radius equal to the inside lap plus the width of steam port, in the points W_1 and H_1. The steam port is then wide open to exhaust while the crank is moving from W_1 to H_1. From the above, it will be seen that the periods are all changed by changing the travel, thus admission and exhaust begin sooner and last longer, and expansion and compression begin later and cease sooner.

For convenience, these results are collected in Table I, which shows the effect of changing the laps, travel, and angular advance.

There are, of course, all sorts of combinations that would make up different problems, but they can all be solved in the same general way, as they are modifications of the problems solved above.

DESIGN OF SLIDE VALVE

In designing a slide valve, some of the variables are assumed and the others are found by means of the diagrams presented above. These diagrams show only the dimensions of the inside and outside laps and travel of valve; the other dimensions of the valve and seat must be calculated.

Area of Steam Port. *Steam Supply Pipe.* It is generally conceded by authorities that the pipes supplying steam to steam engines should be of such dimensions that the mean velocity of steam in them would not exceed 6,000 feet per minute. If the velocity of steam exceeds 6,000 feet per minute, there will be a very appreciable

loss of pressure, which is objectionable. In computing the size of a steam supply pipe for an engine, the assumption is made that the cylinder is filled at each stroke. The volume of steam passing through the steam pipe must equal the total volume of steam used by the cylinder.

Let d equal diameter of steam pipe in inches; D equal diameter of cylinder in inches; L equal length of stroke in feet; and N equal revolutions per minute (r. p. m.).

The area of the steam pipe in square feet would be $\frac{\pi d^2}{4\times 144}$ and that of the cylinder would be $\frac{\pi D^2}{4\times 144}$. The total volume of steam flowing through the pipe per minute would be $\frac{\pi d^2}{4\times 144}\times 6000$. Disregarding the volume of the piston rod, the total volume of steam used by the cylinder in one minute would be $\frac{\pi D^2}{4\times 144}\times 2LN$.

Since the volume of steam flowing through the pipe per minute must equal that used by the cylinder in the same time, we can equate the two expressions; that is,

$$\frac{\pi d^2}{4\times 144}\times 6000 = \frac{\pi D^2}{4\times 144}\times 2LN$$

Solving,

$$d^2 = \frac{D^2LN}{3000}$$

$$d = \frac{D\sqrt{LN}}{54.772}$$

Exhaust Pipe. For exhaust pipes, the mean velocity of steam is taken as 4,000 feet per minute. Therefore

$$\frac{\pi d^2}{4\times 144}\times 4000 = \frac{\pi D^2}{4\times 144}\times 2LN$$

Solving,

$$d^2 = \frac{D^2LN}{2000}$$

$$d = \frac{D\sqrt{LN}}{44.721}$$

EXAMPLE. Suppose an engine is 10 inches × 18 inches, and makes 180 revolutions per minute. Determine the diameters of the steam and exhaust pipes.

SOLUTION. Substituting in the equation

$$d = \frac{D\sqrt{LN}}{54.772}$$

gives for the diameter of the steam supply pipe

$$d = \frac{10\sqrt{1.5 \times 180}}{54.772}$$
$$= \frac{164.3}{54.772}$$
$$= 3 \text{ inches}$$

The required diameter of exhaust pipe would be

$$d = \frac{D\sqrt{LN}}{44.721}$$
$$= \frac{10\sqrt{1.5 \times 180}}{44.721}$$
$$= \frac{164.3}{44.721}$$
$$= 3.67 \text{ inches}$$

A 4-inch pipe would probably be used.

In practice different builders use different formulas, but all are derived from the fundamental assumptions made above, with certain constants added for different types of engines. The size of both steam and exhaust pipes required for engines of the same class is not affected in any marked degree by different types of valve gears.

For a very large engine cutting off early, the allowable velocity may be taken as 8,000 feet per minute instead of 6,000 feet.

Width of Steam Port. The port opening at admission should give nearly as great an area as the steam pipe in order to prevent loss of pressure due to wire-drawing, but the actual width of the port should be great enough for the free exhaust of steam. It is well to have the steam port a little larger than the area of the steam pipe, then with a port opening of six-tenths to nine-tenths of the port area for admission and full port opening at exhaust, satisfactory conditions will result.

The length of the ports is usually made about eight-tenths the diameter of the cylinder. Then in the 10-inch × 18-inch engine, the steam ports would be $.8 \times 10$, or 8 inches long. If the area for admitting steam is 7.0686 square inches (corresponding to a pipe 3 inches in diameter) and the length of port is 8 inches, the width will be $\frac{7.0686}{8}$, or .8836 inch—about $\frac{7}{8}$ inch.

The width of port opening would be about $.9 \times .8836$, or .79524 inch—about $\frac{13}{16}$ inch.

Width of Exhaust Port. When the slide valve is at its maximum displacement, the valve overlapping the exhaust port, as shown in Fig. 7, reduces the area more or less. In designing the valve, the exhaust port should be of such a width that the maximum displacement of the valve does not reduce the area of the exhaust port to less than the area of the steam port. It is not advisable to make the exhaust port too large, for this increases the size of the valve and thus causes excessive friction.

The height of the exhaust cavity should never be less than the width of the steam port and may be made much higher to advantage.

Width of Bridge. The bridge must be of sufficient width so that the outside edges of the valve can not uncover the exhaust port. The width of the steam port plus the width of the outside lap plus the width of the bridge must be greater than the maximum displacement.

The width of the bridges should not be less than the thickness of the cylinder wall in order to make a good casting.

Point of Cut-Off. In the study of Steam Engine Indicators, it was shown that if the point of cut-off is too early, the other events are not good. If a plain slide valve is used with an automatic cut-off, the point of cut-off is controlled either by changing the eccentricity or by changing the angular advance. Either of these methods will accomplish the result at the expense of the compression, which at a very early cut-off may be excessive. Except for locomotives and high-speed engines, where compression is an advantage, the plain slide valve is not arranged to cut off earlier than one-half or two-thirds stroke. If an earlier cut-off is desired, large outside laps are necessary.

Lead. The lead of stationary engines varies from zero to $\frac{3}{8}$ inch according to the style of engine and type of valve gear. An engine having high compression that compresses the steam nearly to boiler pressure will give good results with little or no lead. If the ports are small and the clearance large, there should be considerable lead in order to insure full initial pressure on the piston at the beginning of the stroke. Valves that open slowly require more lead than quick-acting valves.

ILLUSTRATIVE PROBLEM

EXAMPLE. Design and lay out the valve and valve seat for an engine of cylinder diameter 10 inches, stroke 18 inches, revolutions 180 per minute, lead angle 3 degrees, cut-off equal at both ends and taking place at 75 per cent of stroke, maximum port opening .9 area of steam pipe, compression 15 per cent of the stroke at both ends, and length of connecting rod 3 feet.

SOLUTION. The piston displacement, or cylinder volume, will be $\frac{3.1416 \times 10^2}{4} \times 18 = 1413.7$ cubic inches, or .818 cubic feet.

If the engine makes 180 revolutions, neglecting the volume of the piston rod, it will use $2 \times 180 \times .818 = 294.48$ cubic feet of steam per minute. Steam pipe area $= \frac{294.48}{6000} = .0491$ square feet, or 7.07 square inches.

This 7.07 square inches would also be the least possible area of the steam ports. If the length of port is made eight-tenths the diameter of cylinder, the width will be $\frac{7.07}{8} = .88$ inch, or about $\frac{7}{8}$ inch. The width of maximum port opening will be $.9 \times .88 = .792$ inch, or nearly $\frac{13}{16}$ inch.

Zeuner Diagram. It will be necessary to draw a separate valve circle for each end of the cylinder. First, consider the head end. The valve travel not being known, we shall lay off $X\ Y$ on an assumption of 6 inches travel and draw the eccentric circle as shown in Fig. 32. Lay off the lead angle $X\ O\ A_1 =$ 3 degrees. Lay off $X\ C_2 = .75$ of the assumed valve travel $4\frac{1}{2}$ inches. Draw the arc C_1C_2, as previously explained, and draw $O\ C_1$ which will be the crank position at the point of cut-off. The radius of the arc C_1C_2 will be equal to four times the radius of the eccentric circle, or 12 inches, because the connecting rod is four times the length of the crank. Let the line $O\ E_1$ bisect the angle $A_1O\ C_1$, and on $O\ E_1$ draw the valve circle. $O\ V_1$ $(=O\ K_1)$ is then the outside lap, with these assumed conditions. Drawing the lap circle, the maximum port opening E_1N_1 is found to equal $1\frac{7}{16}$ inches, although $\frac{13}{16}$ is all that is necessary. The assumed eccentricity is 3 inches, therefore the probable eccentricity is found from the proportion

$$x : 3 :: \tfrac{13}{16} : 1\tfrac{7}{16}$$

$$x = 1\tfrac{11}{16} \text{ inches}$$

Now draw a new eccentric circle with a radius of $1\frac{11}{16}$ inches and a new valve circle with a diameter $O\ E_2$ $1\frac{11}{16}$ inches. $O\ K_2$ is now the outside lap and

the maximum port opening is equal to $E_2 N_2$, which from actual measurement is found to be $\frac{11}{16}$ inch. The outside lap $O K_2$ ($=O V_2$) is $\frac{23}{32}$ inch and the lead $I J$ is $\frac{3}{32}$ inch.

Produce E_1O to F and draw another valve circle. We shall use this valve circle to determine the outside laps and lead for the crank end of the cylinder. Since the cut-off is to be .75 of the stroke, we may lay off $O H_2 = O C_2$ and, with a radius of 12 inches, draw the arc H_1H_2. Then, as already explained, $O H_1$

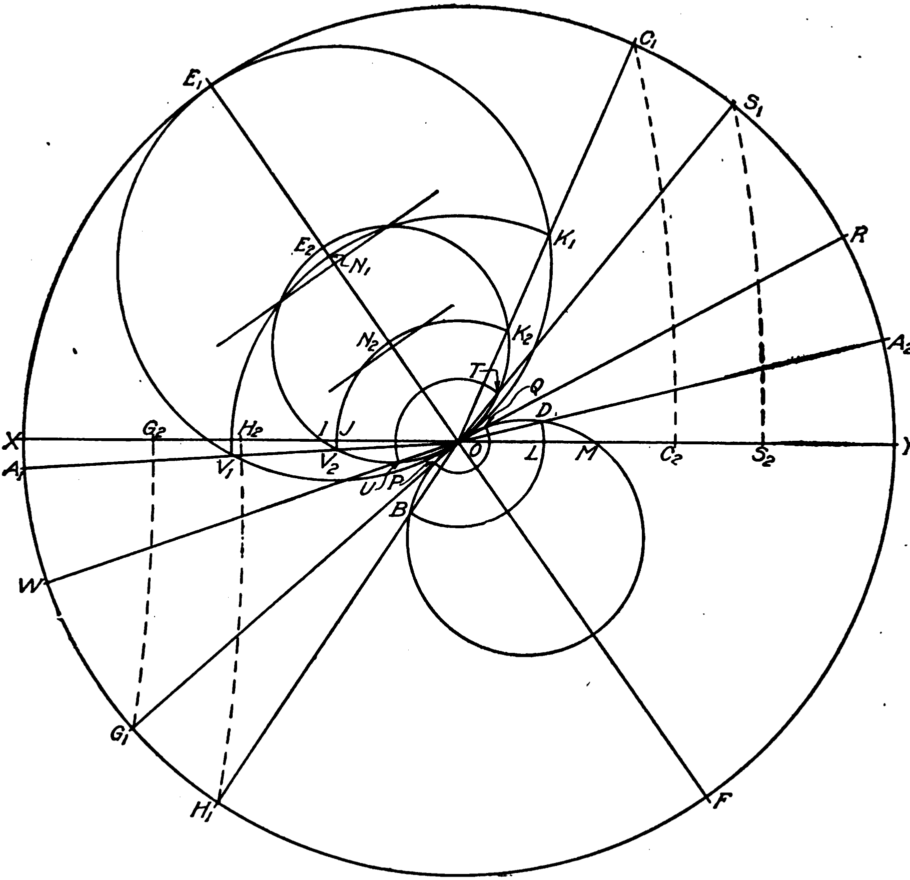


Fig. 32. Zeuner Diagram for Design of Valve and Valve Seat in Problem Page 35

will be the crank angle at cut-off on the return stroke. $O B$, the outside lap, will be $\frac{19}{32}$ inch. Draw the lap circle intersecting the valve circle at D. Then $O D A_2$ is the crank position at admission on the return stroke and $L M$, $\frac{3}{8}$ inch is the lead on the crank end of the cylinder. The maximum port opening will always be greater at the crank end than at the head end because the crank end lap is less in order to get the equal cut-off. If the laps were equal, of course the port openings would be equal.

Now lay off $X\,G_2$ equal to fifteen-hundredths of $X\,Y$ and find the crank position $O\,G_1$. This is the compression on the head end of the cylinder and gives an inside lap on this end of $\frac{7}{32}$ inch, which is equal to $O\,P$. Draw the lap circle $P\,Q$, which allows us to draw through Q the crank line $O\,R$, which is the release on the forward stroke.

Lay off $Y\,S_2$ $(=X\,G_2)$ equal to fifteen-hundredths of $X\,Y$, and construct the crank line $O\,S_1$, which is the crank position at the crank-end compression. $O\,S_1$ intersects the valve circle at T, giving $O\,T$, $\frac{7}{16}$ inch, as the inside lap on the crank end. Draw this lap circle, which will intersect the valve circle at U. This enables us to draw $O\,U\,W$, the crank position at release, on the return stroke.

Layout of Valve. From the data determined by means of these diagrams, the valve may now be laid out. For convenience let us tabulate the results obtained as follows:

Data	Head End	Crank End
Cut-off (per cent of stroke)	75 per cent	75 per cent
Outside lap	$\frac{27}{32}$ inch	$\frac{19}{32}$ inch
Inside lap	$\frac{7}{32}$ inch	$\frac{7}{16}$ inch
Lead	$\frac{3}{32}$ inch	$\frac{3}{8}$ inch
Port opening	$\frac{13}{16}$ inch	$1\frac{1}{16}$ inches
Width of port	$\frac{7}{8}$ inch	$\frac{7}{8}$ inch

Fig. 33 shows this valve in section. Let us begin at the end having the largest inside lap or, in this case, at the crank end. Lay out the steam port $\frac{7}{8}$ inch wide and the crank-end outside lap $\frac{19}{32}$ inch. The bridge will be, say, $\frac{3}{8}$ inch wide. From the inner edge of the steam port, lay off the crank-end inside lap $\frac{7}{16}$ inch. When the valve moves to the left, the point E_2 will travel $1\frac{11}{16}$ inches—a distance equal to the eccentricity—and in this position of extreme displacement, the exhaust port E_1F must be open an amount at least equal to the steam port, $\frac{7}{8}$ inch. Therefore, we lay off $E_1\,F$ equal to $1\frac{11}{16}$ inches $+$ $\frac{7}{8}$ inch $=2\frac{9}{16}$ inches. The inside lap overlaps the bridge nearly $\frac{1}{8}$ inch, so that we shall have to make the exhaust port opening equal to $2\frac{5}{8}$ inches. Lay off $\frac{3}{8}$ inch again for the bridge and measure back $\frac{7}{32}$ inch, equal to the head-end inside lap. The port is $\frac{7}{8}$ inch wide, and the head-end outside lap of $\frac{27}{32}$ inch completes the outline of the valve seat.

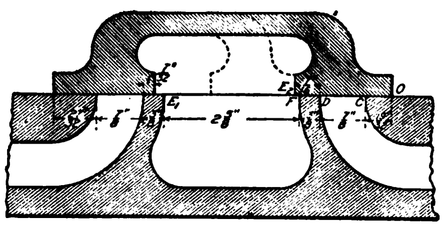


Fig. 33. Section of Valve Designed from Diagram Fig. 32

Reversing Simple Engine. In the operation of a simple engine having a plain slide valve or a piston valve, it sometimes becomes necessary to reverse the direction of rotation of the engine shaft. Remembering the principles presented in the foregoing study of the Zeuner diagram, this is not a difficult task.

It is proposed to here show *first,* how an engine may be reversed with a direct valve, engine running over; *second,* with a direct valve, engine running under; *third,* with an indirect valve, engine running over; and *fourth,* with an indirect valve, engine running under.

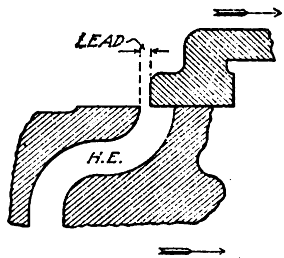


Fig. 34. Section Showing Lead of Valve, Engine Running Over

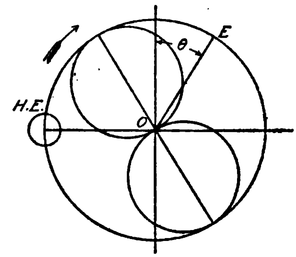


Fig. 35. Diagram for Direct Valve, Engine Running Over

Definitions. Before explaining the operation for obtaining the above, it is well to have an understanding of the meaning of the terms "direct" and "indirect" as applied to a valve, and of "running over" and "under" as applied to an engine.

A valve is said to be a *direct,* or *outside admission, valve,* when at the beginning of the stroke the valve and the piston are moving in the same direction, as indicated by the arrows in Fig. 6. It is

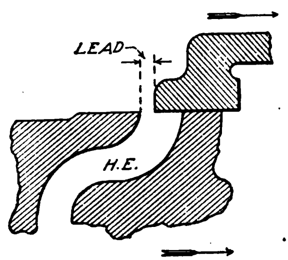


Fig. 36. Section Showing Lead for Direct Valve, Engine Running Under

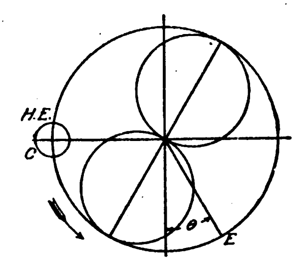


Fig. 37. Diagram for Direct Valve, Engine Running Under

also to be noted that steam is being admitted to the cylinder by the outer edge of the valve, which is the reason for calling it an outside admission valve.

If, in Fig. 6, the valve should be moving in the opposite direction from that shown and steam should be entering the cylinder by the inner edge of the valve, the valve would then be said to be an *indirect,* or *inside admission, valve.*

Most plain slide valves are of the outside admission type, while most piston valves are of the inside admission type.

An engine is said to be *running over*, if, when the piston is moving from the head end toward the crank end, the moving parts, such as connecting rod, crank, etc., are above the center line, as shown in Fig. 17. The engine is said to be *running under* when the above mentioned parts are below the center line when the piston is moving from the head end toward the crank end.

Direct Valve, Engine Running Over. In Fig. 34, let the valve have lead equal to that shown. Since this is a direct valve, engine running over, the valve will be to the right of its mid-position and moving to the right, hence the eccentric will be $(90+\theta)$ degrees ahead of the crank. If the engine is on the head-end dead center, the eccentric would be at E, that is, $(90+\theta)$ degrees ahead of the crank. The right and left valve circles will be located in the second and fourth quadrants, respectively, as shown in Fig. 35.

Direct Valve, Engine Running Under. With a direct valve, engine having lead and running under, as illustrated in Figs. 36 and 37, the valve will be in the same relative position as in the former case, when the crank is on the head-end dead center. In this position the valve must be to the right of its mid-position and moving towards the right, hence the eccentric must be, as shown at E, Fig. 37, an angular distance of $(90+\theta)$ degrees ahead of the crank.

The right and left valve circles will be located in the first and third quadrants, respectively, as shown in Fig. 37.

It is to be noted on comparing the position of the eccentric in Figs. 35 and 37 that both of the eccentric positions make an angle equal to the angle of advance with the vertical. Therefore, to reverse a direct valve, engine running over, turn the eccentric around the shaft, in the direction in which the engine is running, by an angle of $(180-2\theta)$ degrees, or turn the eccentric ahead of the crank, in the direction in which the engine is to run, an angle of $(90+\theta)$ degrees.

Indirect Valve, Engine Running Over. An indirect valve engine running over is illustrated in Figs. 38 and 39. Remembering that the valve must be moving to the left as the piston moves from the head end toward the crank end, and that the valve must be displaced by an amount equal to the lap plus the lead to the left of its mid-position, the eccentric must be below the horizontal and behind

the crank an angular distance of $(90-\theta)$ degrees. Hence, it is located at E, Fig. 39. The right and left valve circles will be located in the fourth and second quadrants, respectively

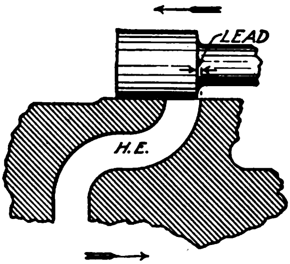


Fig. 38. Section of Indirect Valve, Engine Running Over

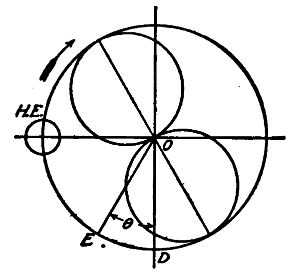


Fig. 39. Diagram of Indirect Valve, Engine Running Over

Indirect Valve, Engine Running Under. To locate the eccentric for an indirect valve engine having lead and running under (see Figs. 40 and 41), proceed as before. The eccentric will be found at E, Fig. 41, and the right and left valve circles will be located in the first and third quadrants, respectively.

An examination of Figs. 38 to 41 will disclose the fact that to reverse an engine using an indirect valve, it is only necessary to turn the eccentric through an angle of $(180-2\theta)$ degrees in the direction in which the engine shaft is turning or, in other words, the procedure is the same as for a direct valve.

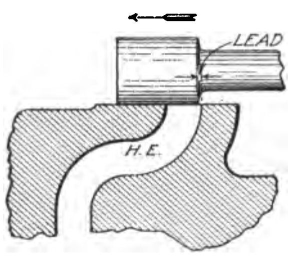


Fig. 40. Section of Indirect Valve, Engine Running Under

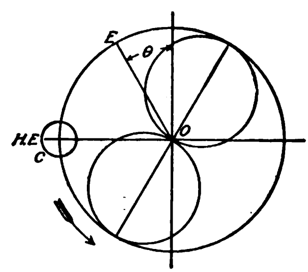


Fig. 41. Diagram for Indirect Valve, Engine Running Under

Comparisons and Comments. A comparison of Figs. 34 and 35 with 38 and 39 will indicate the relative positions of the eccentric for an engine running over with a direct valve and for one running over with an indirect valve. It is evident that in the first case, the eccentric precedes the crank by an angle of $(90+\theta)$ degrees, whereas

in the second, the eccentric follows the crank by an angle of $(90-\theta)$ degrees. This same condition is true for two engines running under, one using a direct valve and the other an indirect.

As an aid in locating the valve travel circles after the eccentric position has been determined, remember that the quadrant separated by a vertical line through the center, from the quadrant containing the eccentric position, is the quadrant in which the right valve travel circle is to be located.

All of the study on the Zeuner valve diagram thus far has to do with an engine running over having a direct valve. After the location of the eccentric position has been determined for the above various conditions, the construction of the Zeuner diagram should be a simple matter.

The principles underlying the location of the eccentric for an engine running over or under and having a direct or indirect valve should be borne in mind when setting valves.

VALVE SETTING

Possible Adjustments. The principles of valve diagrams are useful in setting valves as well as in designing them. The valve is usually set as accurately as possible, and then, after indicator cards have been taken, the final adjustment can be made to correct slight irregularities.

The slide valve is so designed that the laps can not be altered without considerable labor, and the throw or eccentricity of the eccentric, which determines the travel of the valve, is usually fixed. The adjustable parts are commonly the length of the valve spindle and the angular advance of the eccentric.

By lengthening or shortening the valve spindle, the valve is made to travel an equal distance each side of the mid-position. Moving the eccentric on the shaft makes the action of the valve earlier or later as the angular advance is increased or decreased.

To Put Engine on Center. It is usual to put the engine on center before setting the valve. First, put the engine in a position where the piston has nearly completed the outward stroke and make a mark M_1, Fig. 42, on the guide opposite the corner of the crosshead at some convenient place. Also make a mark P with a center punch on the frame of the engine near the crank disk. With

this mark P as a center, describe an arc C on the wheel rim with a tram.*

Turn the engine past the center until the mark on the guide again corresponds with the corner of the crosshead and make another

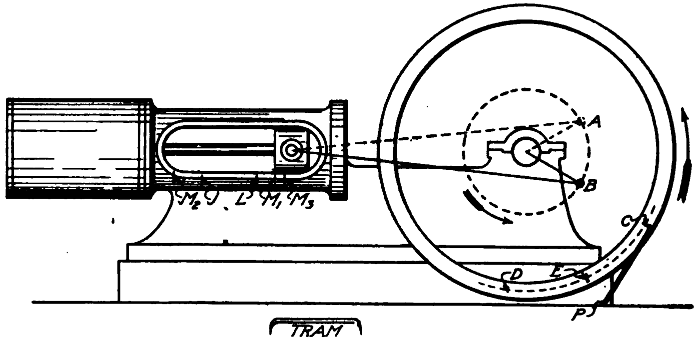


Fig. 42. Sketch of Engine, Showing Method of Putting Engine on Center

mark D on the wheel with the tram, keeping the same center. With the center of the pulley, or crank disk, as a center, describe an arc CD on the rim, which intersects the two arcs drawn with the tram. Bisect the arc CD and turn the engine until the new point is distant from the point P an amount equal to the length of the tram, in which position the engine will be on center.

The engine should always be moved in the direction in which it is to run so that the lost motion of the wrist pin and crank pin

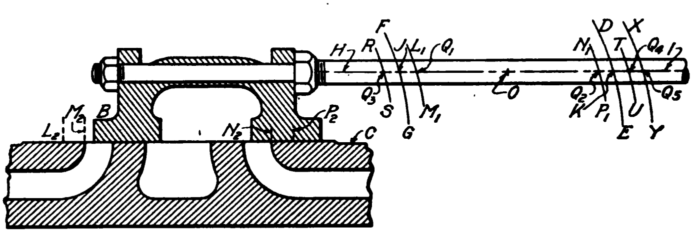


Fig. 43. Diagram Showing How Valve is Set for Equal Lead

will be taken up the right way. In case the engine has been moved too far at any time, it should be turned back beyond the desired point while the engine is moving in the proper direction. In this manner, the dead center can be located for both the head and crank ends.

*A tram is a steel rod with its ends bent at right angles and sharpened.

To Set Valve for Equal Lead. After locating the dead center points as described above the next step is to locate what are known as the port marks. In Fig. 43 move the valve to the left until cut-off occurs on the head end or until the edge of the valve at B is at M_2. Then, with a center C on some fixed point on the cylinder or engine frame, describe with a tram the arc $F\ G$ on the valve rod. Continue the rotation of the engine in the same direction until cut-off takes place at the crank end. Then with the same tram and center C, sweep the arc $D\ E$ on the valve rod. Draw the center line $H\ I$ and where this center line cuts the arcs $F\ G$ and $D\ E$, mark the points J and K, respectively, which points are known as the port marks. Bisect the distance between J and K, thus establishing the point O. When one tram point is in C and the other just enters the point J, the valve is just cutting off on the head end; and when the tram point coincides with C and K, it is an indication that cut-off is occurring on the crank end, hence a basis of comparison has been established for the two ends. Place the engine on the forward dead center and sweep the arc L_1M_1. The distance between the arcs L_1M_1 and $F\ G$, which is equal to JQ_1, represents the amount the valve extends over the port when the engine is on the head-end dead center. In a like manner, establish the arc N_1P_1 when the engine is on the crank-end dead center, in which position the valve overlaps the steam port the distance KQ_2. In order to have equal travel of the valve on either side of its mid-position, the distance $J\,Q_1$ should equal $K\,Q_2$. If necessary to equalize these distances, lengthen or shorten the valve stem as required. Having secured an equal valve travel, place the engine on the forward dead center. Since the engine is running under (see Fig. 37), the eccentric will be found $(90 + \theta)$ degrees ahead of the crank in the direction the engine is to run. Lay off on the valve stem the distances $J\,Q_3$ and $K\,Q_4$ equal to the required lead. With the tram point in C and the engine placed on the head-end dead center, turn the eccentric in the direction in which the engine is to run—which is *under* in this case—until the arc $R\ S$ passes through the point Q_3. Fasten the eccentric and turn the engine around until it is on the crank-end dead center. Sweep another arc as $T\ U$ with the tram. If this arc passes through the point Q_4, then the valve is correctly set for equal lead, that is, JQ_3 is equal to KQ_4. If, however, the arc $T\ U$

does not pass through the required point Q_4, but falls beyond, it is an indication of unequal lead, so a correction must be made. Suppose, for instance, that when the crank was placed on the crank-end dead center, the arc described from C fell at XY instead of TU, then it is obvious that the crank end has more lead than the head end. To make a correction for this inequality, find the difference between the lead on the head and crank ends—which in this case is equal to the distance Q_4Q_5—and correct half of the difference on the valve stem and the other half by altering the angle of advance. In this case, the valve stem should be lengthened by the amount $\frac{Q_4Q_5}{2}$, which would increase the lead on the head end by that amount and decrease it by the same amount on the crank end. After establishing an equal travel of the valve by adjusting the length of the valve stem, thus giving an equal amount of lead at each end, the desired amount of lead may be obtained by changing the angle of advance. To obtain the required lead in this case, it would be necessary to reduce the angle of advance. It may be necessary to make several trials before the desired results are obtained, this being particularly true if working on an engine having lost motion in the various parts. In order to eliminate the effect of lost motion in so far as possible, the engine should always be turned in the direction which it is to run.

In case it is difficult to turn an engine, the following method may be used. First, loosen the eccentric on the shaft and turn it around until it gives a maximum port opening first at one end and then at the other. If the maximum port openings are not equal, make them so by changing the length of the valve spindle by half the difference. When the above adjustment has been made, set the engine on dead center and give the valve the proper lead by turning the eccentric on the shaft. The angular advance is thus adjusted.

To Set Valve for Equal Cut-Off. To set the valve for equal cut-off, observe the preliminary steps of locating on the valve stem the dead-center points, port marks, and equal travel of the valve to either side of its mid-position, as described in connection with setting the valves for equal lead.

Assume that it is desired to set the valves for an equal cut-off of 75 per cent. On the guides of the engine illustrated in Fig. 42,

locate the points M_2 and M_3, corresponding to the extreme positions of the edge of the crosshead, or a given point on the crosshead. The distance M_2M_3 represents the stroke of the piston, so when 75 per cent cut-off occurs, the reference point on the crosshead should be at a point J, which is 75 per cent of the stroke M_2M_3 for the crank end and at the point L for 75 per cent cut-off on the head end. Remembering that the points J and K on the valve stem in Fig. 43 represent points of cut-off, all required reference points needed are known. Turn the engine over in the direction indicated in Fig. 42 until the reference point on the crosshead corresponds to the reference point on the guide, as L, for the head-end cut-off. Then with the tram in the center C, Fig. 43, describe an arc, say, L_1M_1. Continue the rotation of the engine in the same direction until the piston has completed the forward stroke and has returned to the point where the reference lines on the crosshead and the guide J coincide. Tram the valve stem as before, locating the arc, say, N_1P_1. Since the tram should coincide with the arcs $F\,G$ and $D\,E$ for the head-end and crank-end cut-off, respectively, it is therefore evident that with the tram coinciding with L_1M_1 and N_1P_1 that the required cut-off is not obtained but occurs too early. Since the distances Q_1J and Q_2K are equal, the length of the valve stem does not need to be disturbed. To make cut-off occur later, decrease the angle of advance by moving the eccentric opposite to the direction in which the engine is to run. For instance, with the engine standing so that the point L, Fig. 42, and the end of the crosshead are coincident, move the eccentric until the tram points coincide with C and J, Fig. 43. Try the points for the cut-off on the crank end, and if the tram fits easily into C and K, then the valve is set correctly. If, however, the tram points do not fit into the points C and K, continue the operation until the desired results are obtained.

From the above discussion, two points have been established:

(1) Moving the valve on the valve rod changes the corresponding events the same on both ends, one being made earlier and the other later. That is, if the cut-off is made earlier on the head end, it will be later on the crank end, and so on for the other events.

(2) Moving the eccentric on the shaft or changing the angle of advance changes the corresponding events the same for both ends, both being made earlier or both later.

MODIFICATIONS OF THE SLIDE VALVE

Balancing Steam Pressure. The ordinary slide valve is most suitable for small engines. For engines of large size, some method must be employed to balance the steam pressure on the back of the valve. With large valves, such for instance as those of locomotives or large marine engines, a great force is exerted by the steam, and the valve is forced against its seat so hard that a large amount of

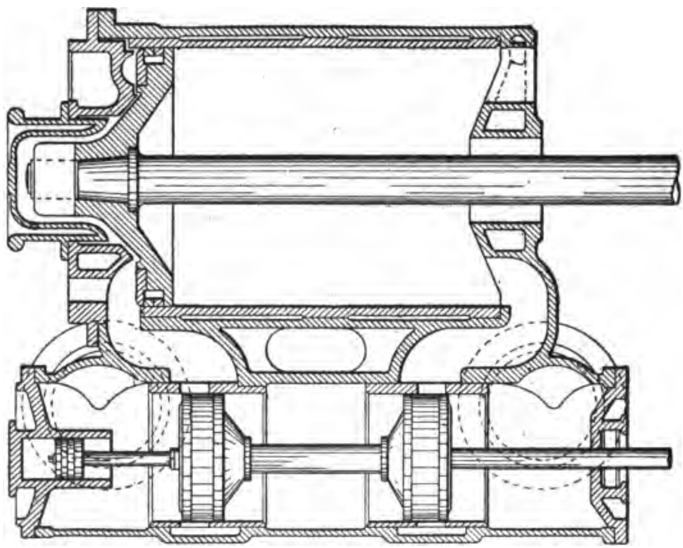


Fig. 44. Section of Piston Valve and High-Pressure Cylinder of U. S. S. "Massachusetts" Showing Method of Balancing

power is necessary to move it. This excessive pressure causes the valve to wear badly and is a dead loss to the engine. The larger the valve, the greater this loss will be.

Piston Valve. To prevent excessive pressure on the back of the valve, the piston valve is commonly used, especially in marine engines. This valve consists of two pistons which cover and uncover the ports in precisely the same manner as the laps of the plain slide valve. These pistons are secured to the valve stem in an approved manner and are fitted with packing rings.

The valve seat consists of two short cylinders or tubes accurately bored to fit the pistons of the valve. The port openings are not continuous as in the plain slide valve, but consist of many small openings, the bars of metal between these openings preventing the packing rings from springing out into the ports.

Steam may be admitted to the middle of the steam chest and exhausted from the ends or *vice versa.* With the former method, the live steam is well separated from the exhaust, and the valve-rod stuffing box is exposed to exhaust steam only. This is a good arrangement for the high-pressure cylinder; if used for a cylinder in which there is a vacuum, air may leak into the exhaust space through the valve-rod stuffing box. With this arrangement, the steam laps must be inside and the exhaust laps on the outside ends.

The piston valve may be laid out and designed by means of the Zeuner diagram just as if it were a plain slide valve, and the action

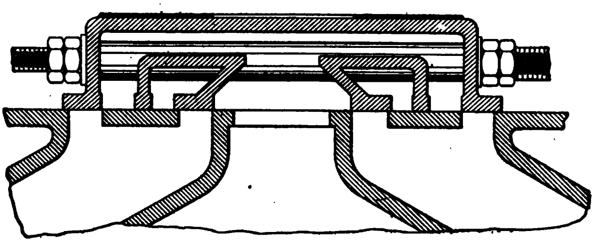


Fig. 45. Section of Double-Ported Slide Valve

is the same except that it is balanced so far as the steam pressure is concerned, the power to drive it being only that necessary to overcome the friction due to the spring rings.

Fig. 44 shows a section of the piston valve and the high-pressure cylinder for one of the engines of the U. S. S. "Massachusetts." This valve consists of two pistons connected by a sleeve through which the valve rod passes. This valve rod is prolonged to a small balancing piston, placed directly over the main valve. The upper end of the balancing cylinder does not admit steam, so that the steam pressure below the balancing piston will practically carry the weight of the piston valve, thus relieving the valve gear and making the balance more nearly complete.

Double-Ported Valve. Sometimes it is impossible to get sufficient port opening for engines of large diameter and short stroke,

especially those having a plain slide valve with short travel. This difficulty may be overcome by means of the double-ported valve shown in Fig. 45. It is equivalent to two plain slide valves, each having its laps. The inner valve is similar to a plain slide valve except there is communication between its exhaust space and the exhaust space of the outer valve. Each passage to the cylinder has two ports; a bridge separates the exhaust of the outer valve from the steam space of the inner valve, and the outer valve is made long enough to admit steam to the inner valve.

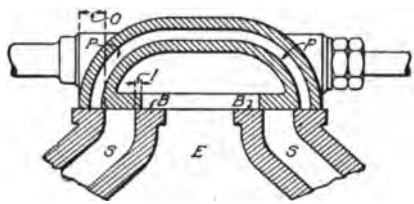


Fig. 46. Trick Valve Shown in Mid-Position

This valve may be considered as equivalent to two equal slide valves of the same travel, each having one-half the total port opening. To admit the same amount of steam as a plain slide valve, the double-ported valve requires but half the valve travel; this is advantageous in high-speed engines.

To balance the excessive steam pressure, the back of the valve is sometimes provided with a projecting ring which is fitted to a similar ring within the top of the valve chest. These rings are planed true and fit so that steam is prevented from acting on the back of the valve.

Trick Valve. The defect of the plain slide valve, due to the slowness in opening and closing, is largely remedied in the trick valve, which is so made that a double volume of steam enters during admission. Thus a quick and full opening of the port is obtained with a small valve travel.

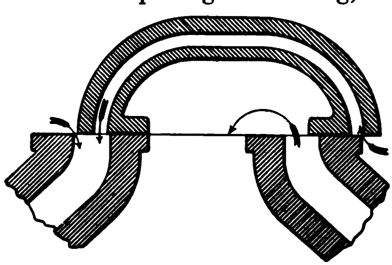


Fig. 47. Trick Valve Showing Admission of Steam Just Beginning

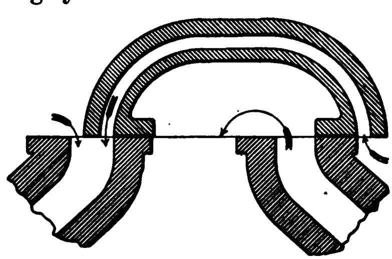


Fig. 48. Trick Valve at Extreme Right Position with Steam Port Open Wide

In Fig. 46 the valve is shown in mid-position. It is similar to a plain slide valve except that there is a passage *P P* through it. It has an outside lap *O* and an inside lap *I*. The seat is raised and has steam ports *S S*, bridges *B B*, and exhaust port *E*. If the valve moves to the right a distance equal to the outside lap plus the lead, it will be in the position shown in Fig. 47. Steam will be admitted at the extreme left edge of the valve just the same as though it were a plain slide valve; also, since steam surrounds the valve, it will be admitted through the passage as shown in Fig. 47. If the lead is the same as for a plain slide valve, $\frac{1}{16}$ inch for instance, this valve would give double the port opening, that is, $\frac{1}{8}$ inch, when the valve was open a distance equal to the lead.

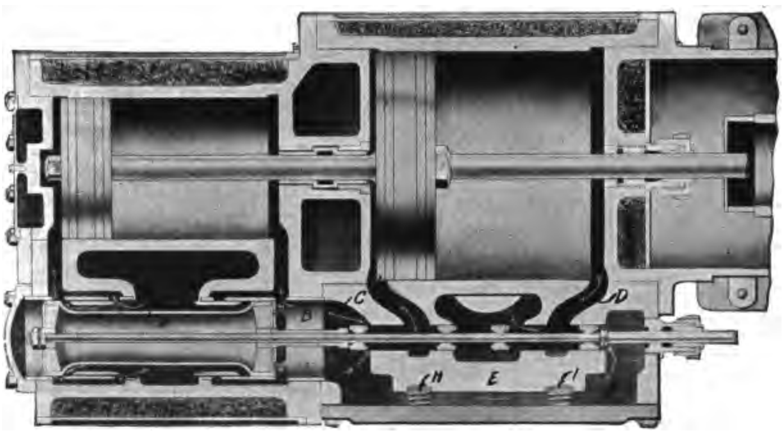


Fig. 49. Obtaining Perfect Balance by Use of Double-Ported Piston Valve and Double-Ported Slide Valve in Compound Engine

Fig. 48 shows the valve when it is in extreme position to the right and the port is full open to steam.

Piston valves are also made with a passage similar to that of the trick valve for double admission, that used with the Armington and Sims engine being, perhaps, the best example.

Application of Various Types. Piston valves are commonly used on the high and intermediate cylinders of triple-expansion engines, and if well made and fitted with spring rings, should not leak. Small piston valves are often made without packing rings; but even if they fit accurately when new, they soon become worn and cause trouble.

The double-ported valve, the trick valve, and others, often have some device for relieving the pressure, such as a bronze ring or cylinder fastened to the back of the valve. This ring is pressed by springs against a finished surface of the valve chest cover, and the space thus enclosed by the ring may be connected to the exhaust. There are numerous devices for balancing valves, but they are usually more or less expensive and are liable to cause trouble from leakage.

Fig. 49 well illustrates the application of a double-ported piston valve and a double-ported slide valve to a compound engine. It also shows a method used for obtaining a perfect balance. The piston valve on this engine is a hollow inside admission valve. The steam passes from the cavity *A* through the double ports in the piston valve, forcing the high pressure piston to the right, which action causes the exhaust steam to pass out of the high pressure cylinder through the passage *B* into the steam chest of the low pressure cylinder. The steam passes around the flat valve at *C C* into the low pressure cylinder. The steam back of the low pressure piston passes through the port *D* into the exhaust cavity. The pressure plate *E* is held against the flat slide valve by the springs *H* and *I*, there being steam all around the pressure plate, as at *F* and *G*. The valve fits closely between the valve seat and pressure plate, but the pressure plate being supported at the sides eliminates the pressure between the valve and its seat. Both of these valves are said by the builders to be in perfect balance.

Reversing Mechanism. In the early development of valve gears, it became necessary to devise some means of reversing the engine, hence it is found that a great many of the most prominent gears, such as the Stephenson, Walschaert, Marshall, and many others of more or less merit, embody the reversing feature.

Reversing by Means of One Eccentric. At first, the reversing of an engine was accomplished by the use of one eccentric, there being two methods by which this was done.

(1) The device shown in Fig. 50 was used on some of the earliest locomotives and marine engines, and may now be found as the reversing medium for engines used on small launches. The eccentric *E* is loose on the shaft between a fixed collar *G* and a hand wheel *H*. A stud projecting from the eccentric passes through a curved

slot in the disk of the wheel and can be clamped by a hand nut *F*. When running forward with the crank at *C*, the center of the eccentric is at *A* and the nut is clamped at *F*. To reverse, steam is shut off and, when the engine stops, the nut *F* is loosened and then moved to *B* and clamped; or, after *F* is loosened, the wheel, shaft, crank, and propeller are turned over by hand until *B* strikes the stud at *F*, where it is clamped. The engine will then run astern.

(2) The eccentric was mounted on a sleeve, which could be moved longitudinally along the shaft of the engine by means of a lever. The sleeve had a spiral slot cut on the inside of it, which subtended an angle of $(180-2\theta)$ degrees. This slot fitted over a radial pin on the shaft, so when the sleeve was pushed in or out by the lever, both the sleeve and the eccentric were turned through $(180-2\theta)$ degrees, thus reversing the engine.

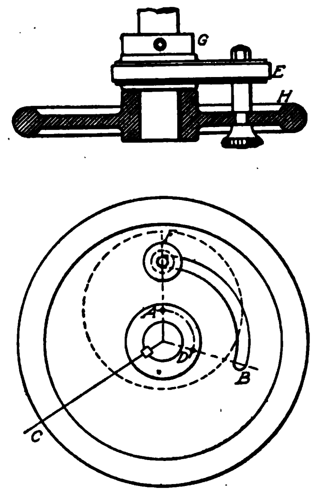


Fig. 50. Early Reversing Device by Means of One Eccentric

Reversing by Means of Two Eccentrics and Gab-Hooks. It is obvious from the foregoing that the method of reversing by shifting one eccentric is awkward and not well adapted to high speeds and large engines. It was a natural transition, therefore, from the one eccentric to the more convenient reversing gears having two eccentrics, one set $(90+\theta)$ degrees ahead of the crank for the forward motion and one $(90+\theta)$ degrees behind the crank for the backward motion. At first, this arrangement was rather crude and objectionable in some respects, as will be noted later. The essential feature to be borne in mind with reference to a two-eccentric gear is, that the object is to have only one eccentric at a time operating the valve. In the early development, this was accomplished by using gabs or gab-hooks, which could be brought in contact with the valve rod at the pleasure of the operator. For instance, if the engineer wished to go forward, he would lower the arm *R*, Fig. 51, thus bringing B_1 in contact with the valve rod at *V*. The valve would then be operated by the forward eccentric E_1 and the engine would run

in the direction indicated by the arrow at E_1. To reverse the engine, B_1 would be disengaged and B_2 placed in connection with V. The valve would then be operated by the eccentric E_2, and the engine would run in the direction indicated by the arrow at E_2, which is the reverse of that indicated at E_1. It is to be particularly noted that only one eccentric actuates the valve at one time. All reversing gears of the two-eccentric type carry out this principle to a greater or less extent.

It will be obvious that the gab-hooks are an improvement over the shifting eccentric, but even they have certain objectionable features, the three principal ones being (1) the engine must have a slow speed of rotation; (2) the engine must be of such construction that

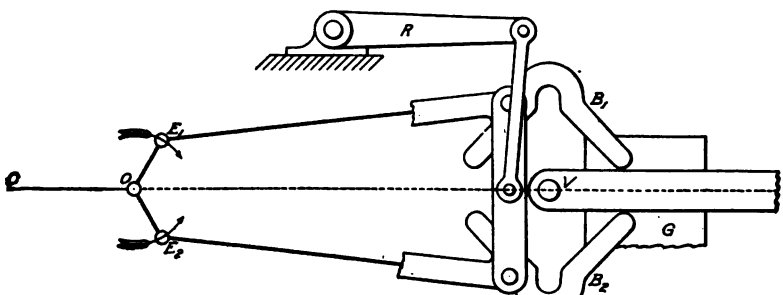


Fig. 51. Reversing Device Using Two Eccentrics and Gab-Hooks

the reversal can be accomplished in a leisurely manner—it is not convenient to reverse at a high speed with a gab-hook, but the engine must be turning slowly when the hook is dropped upon the pin; (3) the engine must be of such a type that it can be started by hand-working of the valves.

Reversing by Means of Two Eccentrics and Curved or Straight Links. To overcome these objectionable features, a step forward was taken when the gab-hooks were replaced by the curved or straight link, which is now used in connection with almost all reversing gears. This was a decided improvement as it not only accomplished the reversing of the engine but also made possible a variation in the adjustment of the valve mechanism, which permitted much more economical distribution of steam in the cylinder. There are two general classes of valve gears that use the curved link and its neces-

sary attachments, namely, the shifting link or the stationary link type, and the radial gear type.

The Stephenson gear is a worthy exponent of the shifting link type. The Walschaert, Joy, Marshall, and others are representatives of the radial gear type.

SHIFTING LINK TYPE OF VALVE GEAR

Stephenson Link Motion. As the Stephenson gear is one of the oldest reversing gears used and is perhaps the best known, a discussion of its principal features is presented first. This gear has been successfully used on stationary, traction, and marine engines, but its largest and, perhaps, most successful application has been on American locomotives. This gear is illustrated in Fig. 52. The two eccentrics E_1 and E_2, whose centers are at C_1 and C_2, respectively, are shown in their relative positions when the crank OA is at the crank-end dead center. The eccentric rods R_1 and R_2 are connected by forked ends to the link pins H and G. The link consists of two curved bars bolted together in such a manner that they may slide by the link block N. On the link are three sets of trunnions; the two outer ones, or link pins, are fitted into the forked end of the eccentric rods, and the middle one, known as the saddle pin, is fitted into the end of the drag links $F M$.

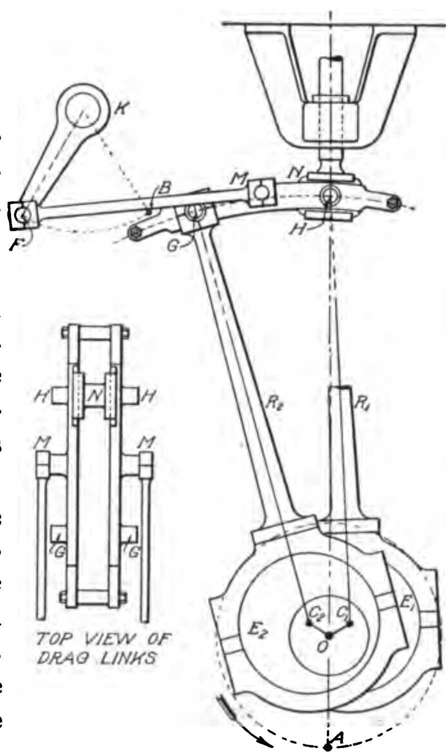


Fig. 52. Stephenson Link Motion

The valve stem has, at its lower end, a pivoted block N, called the *link block,* provided with slotted sides through which the links

can slide. The reverse shaft, or rock shaft, K, here shown in the full gear "forward," may be turned until F moves to B; in this position the link will be pushed across the link block, and the valve will get its motion from the rod R_2 instead of from R_1, as before. The link in this position would be in full gear "backward."

From the foregoing, it is obvious that the Stephenson gear may be divided into three distinct mechanisms, each of which perform a definite function. *First,* the link motion proper, comprising the parts from the axle to the link; *second,* the adjusting gear, which is composed of the lifting shaft and reversing lever by which the power of the engine is controlled by lowering or raising the link; and, *third,* the valve and its attachments. The link motion proper is,

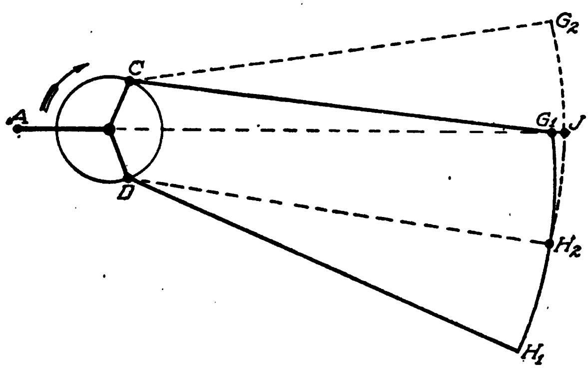


Fig. 53. "Open Rod" Arrangement of Eccentric Rods in Stephenson Gear

perhaps, the most important of the three, at least for the present study. Remembering that the link supplanted the gab-hook, it should be obvious that the eccentric rods and their connection to the link form a combination similar in action to the gab-hooks and valve rod, with some intervening parts which do not materially affect or change the operation.

Relative Position of Eccentric Rods. In order to have clearly in mind just what action does take place when the link is shifted from one position to another, it is essential that the relative position of the eccentric rods be understood. They are designated as "open rods" when arranged as shown in Fig. 53, with the eccentric centers C and D on the same side of the axle as the link, and "crossed rods" when the rods cross as illustrated in Fig. 54. The location, length, and attachment of the eccentric rods to the link have a mate-

rial effect upon the movement of the valve. Experience and calculations have shown that the eccentric rods should not be shorter than eight times the throw of the eccentric. They are usually much longer than this. The distance between the eccentric rod pins should not be less than two and one-half times the throw of the eccentric. If the distance is less than this amount, the angle between the link and the block will be such that there will be an excessive slip of the block and undue stresses in the mechanism will be induced. The angularity of the eccentric rods produces irregularities in the movement of the valve, which can be largely compensated for by locating the saddle pin inside the center line of the arc, but not too far inside for then it would give a long slip of the link and be objectionable. The adjustment of the link also requires that special atten-

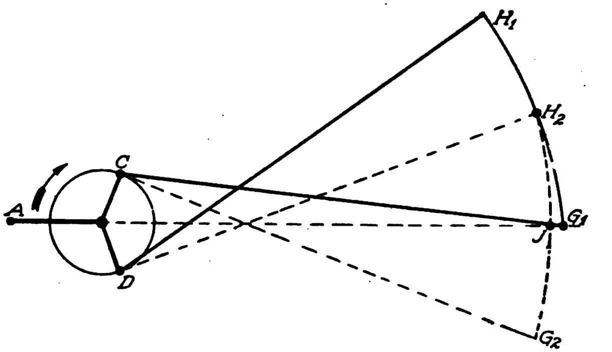


Fig. 54. "Crossed Rod" Arrangement of Eccentric Rods in Stephenson Gear

tion be given to the amount of lead at full gear as well as to the increase of lead produced by "hooking up" the engine. With "open rods" it will be seen that when in full gear the link block is at G_1, and that if, without turning the crank, the link is shifted to mid gear, then the link block moves to J, Fig. 53, and the valve must consequently be moved toward the right an amount equal to G_1J, thereby increasing the lead on the crank end of the cylinder. With "crossed rods," moving the link from full to mid gear moves the link block from G_1 to J, Fig. 54, thus reducing the lead. It follows then that open rods give increasing lead from full toward mid gear, and that crossed rods give decreasing lead. With crossed rods there will be no lead when in mid gear. It will be apparent that the shorter the rods, the greater this increase or decrease will be.

The open rods are more generally used than the crossed rods; especially is this true in locomotive service. The feature of increasing lead from full to mid gear is the distinguishing characteristic of the Stephenson gear. When the engine is in full gear, so that the forward link pin G_1 is on the center line as in Fig. 53, then only the eccentric C controls the valve, and the travel of the valve will be equal to twice the throw of the eccentric C. In other words, when in full gear, only one eccentric moves the valve, as was the case when using gab-hooks. As the link is raised, both of the eccentrics have an effect on the motion of the valve, the result being very much the same as if another controlling eccentric of shorter throw were introduced. The throw of this resultant eccentric would decrease until the center was reached, when it would be a minimum. Finally, the

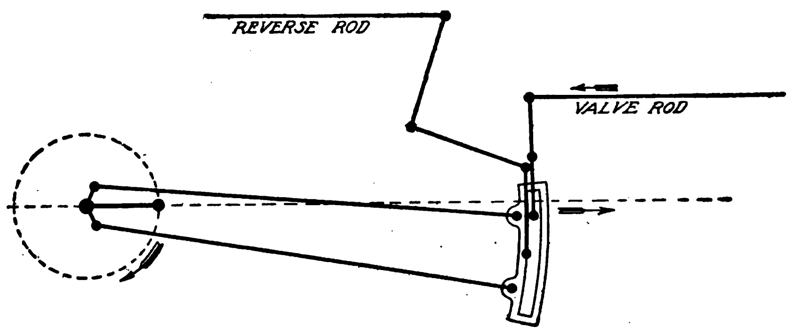


Fig. 55. Diagram of Stephenson Gear Showing Link Block and Rocker

center of this resultant eccentric would be on the center line of the motion, midway between the two actual eccentrics. At this point, the radius of the resultant eccentric would be equal to the sum of the lap and lead in full gear. Therefore, in mid gear, the valve travel is equal to twice the sum of the lap and lead in mid gear.

Location of Link Block. Nearly all marine engines, and some English locomotives, have their link blocks carried directly on the valve rod. American locomotives commonly use a rocker, one end of which carries the link block while the other moves the valve rod. This arrangement, indicated in Fig. 55, makes it possible to place the valve and steam chest above the cylinder. The position of the crank for the same valve position is just opposite that shown in Fig. 53, because the rocker reverses the direction of motion of

the valve. While apparently the crossed rod arrangement is used, yet it is really the open rod arrangement and gives increasing lead toward mid gear. A rod from the bell crank lever on the reverse shaft E leads back to the engineer's cab and connects with the reverse lever. This lever moves over a notched arc and may be held by a latch in any one of the notches, thus setting the link in any position from mid gear to full gear, either forward or back.

The Stephenson link is designed to give equal lead at both ends of the cylinder; but to accomplish this, the radius of the link arc (that is, an imaginary line in the center of the slot) must be equal to the distance from the center of this slot to the center of the eccentric. In Fig. 52, the radius of the link arc is equal to C_1H and C_2G.

Exact equality of lead is not essential, and the radius of the link arc is sometimes made greater or less than stated above in order to aid in equalizing the cut-off; but the change should never be great enough to affect the leads.

Application to Expansion and Cut-Off. Stephenson originally intended to use the link simply as a reversing gear, but soon found, however, that at intermediate points between the two positions of full gear, it would serve very well as a means of varying the expansion and cut-off. Very soon, the link came to be used not only on locomotives and marine engines, but on stationary engines as well, in connection with the reverse shaft which was under the control of the governor. The mechanism proved to be too heavy to be easily moved by a governor and it has gradually fallen into disuse on stationary engines excepting as a means of reversing.

In marine practice, the variable expansion feature is of little value, for marine engines run under a steady load and the link is set either at full gear or at some fixed cut-off. For locomotives, however, the variable expansion is nearly as important as reversing. Locomotives are generally started at full gear, admitting steam for nearly the entire stroke and then exhausting it at relatively high pressure. This wasteful use of steam is necessary to furnish the power needed in starting a train. After the train is under way, less power is required per stroke and the link is gradually moved toward mid gear or "hooked up" by the engineer, thus hastening the cut-off; the expansion is increased and the power is reduced in proportion to the load.

As the cut-off is changed, it is desirable to maintain an approximately equal cut-off at each end of the cylinder; this can be secured in the Stephenson gear by properly locating the saddle pin and the reverse shaft. When used without a rocker, as in Fig. 52, the saddle pin should be on the arc of the link or slightly ahead of it. When used with a rocker, the saddle pin should be behind the link arcs and, in order to give symmetrical action for both forward and backward running, it should be opposite the middle of the arc, that is, equally distant from each link pin.

Zeuner Diagram for Stephenson Gear. The Stephenson link can not be designed directly from the Zeuner diagram, but a systematic investigation can be made by using a wooden model of the proposed link. This can be mounted on a drawing board, and the effect of changing the position of pins and the proportions of rods and levers can be determined without difficulty. By a system of trials, a combination can be found best suited to obtain the desired results. Moreover, a model makes it possible to measure directly the slip of the link block along the link. This slip should be kept as small as possible to prevent rapid wear. It can be controlled to some extent by properly locating the link pins, by avoiding too short a link, and by choosing a favorable position for the reverse shaft.

The Zeuner diagram for a Stephenson gear embodies all of the principles of the Zeuner diagram for a simple valve, with certain additional ones which, while comparatively simple, sometimes cause confusion. It is only necessary to remember that there are two eccentrics and that their combined action is the same as one virtual eccentric; also, that in passing from a long to a short cut-off with open rods, the lead increases, hence the path of the virtual eccentric center must be a curved one. A practical example will make the construction of such a diagram clear.

Example. Given a maximum valve travel of $5\frac{1}{2}$ inches, steam lap 1 inch, lead at full gear $\frac{1}{16}$ inch, and $\frac{R}{L}$ equal to $\frac{1}{4}$. Find the valve travel for 60 and 80 per cent cut-off, respectively.

Solution. Construct the valve travel circle $A\ B\ C\ D$, Fig. 56, having a diameter of $5\frac{1}{2}$ inches. (The scale of the drawing is exactly $\frac{3}{8}$ size.) Draw the lap circle $T\ U\ V\ W$ and lay off the full gear lead $S\ T$. Lay off the angles $P\ O\ B$ and $Q\ O\ D$ equal to the angle through which the eccentrics must be turned in order to displace the valve by an amount equal to the lap plus the lead at full gear; or with slight error draw a perpendicular to $A\ C$ through

the point S and where it cuts the maximum valve circle, as at P and Q, will be the centers of the eccentrics sought. Two points of the locus of the virtual eccentric center have thus been established. In order to draw the locus, the amount of lead at mid gear must be known. By the construction of Fig. 53, it was shown that in shifting the link from the full gear position G_1H_1 to the mid gear position G_2H_2, the lead was increased by the amount G_1J, which can be measured directly from the drawing. In this problem assume that the

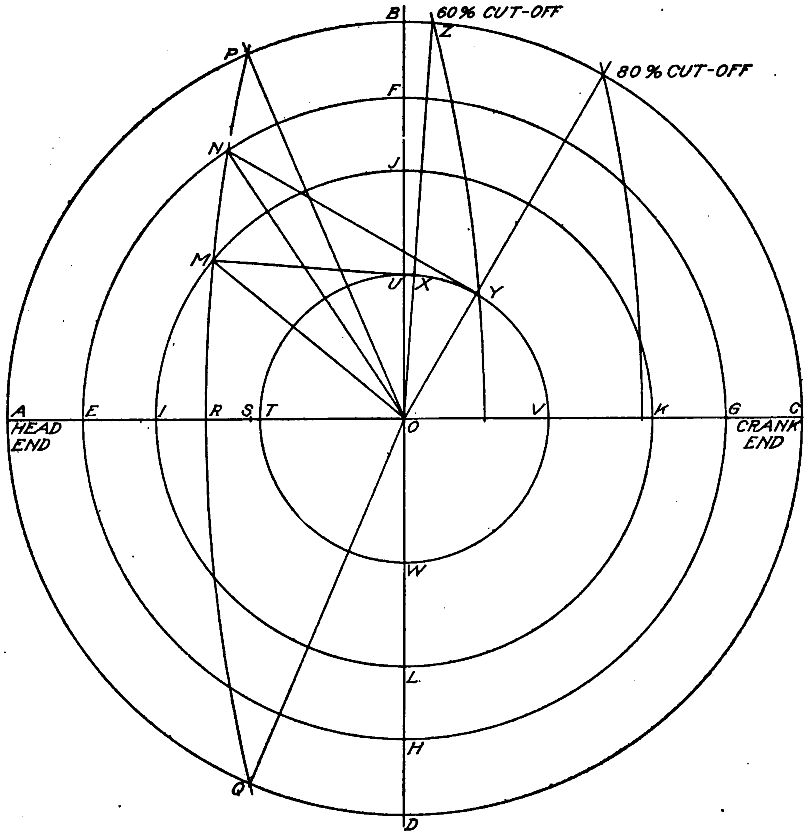


Fig. 56. Zeuner Diagram for Stephenson Gear

length of the eccentric blades is known and that by a construction similar to that in Fig. 53, the lead at mid gear was found to be $\frac{3}{8}$ inch, or, in other words, in passing from full to mid gear the lead was increased $\frac{5}{16}$ inch. Knowing the lead at mid gear, lay off the distance TR equal to $\frac{3}{8}$ inch. The locus of the virtual eccentric center must pass through P, Q, and R and have its center on the line AC extended. By trial, we find such a center and such a radius that the arc when drawn will pass through the points P, Q, and R. This arc is the locus of the virtual eccentric center when dealing with the head-end events. To find the events for the crank end, construct a similar arc on the right of the

vertical line $B D$. To obtain the valve travel for 60 per cent cut-off, first determine the crank position in the usual manner by locating the line $O Z$, remembering that $\frac{R}{L} = \frac{1}{4}$. Where this line cuts the lap circle, as at X, draw a tangent to the lap circle and extend it until it cuts the arc $P R Q$ at M. $O M$ is then the radius of the valve travel circle for 60 per cent cut-off. Construct the valve travel circle $I J K L$ with a diameter of $3\frac{1}{2}$ inches, the required valve travel. In like manner, establish the point N, which determines the valve travel circle $E F G H$ for 80 per cent cut-off, the diameter of which is $4\frac{1}{2}$ inches. By this same procedure, the valve travel for any cut-off

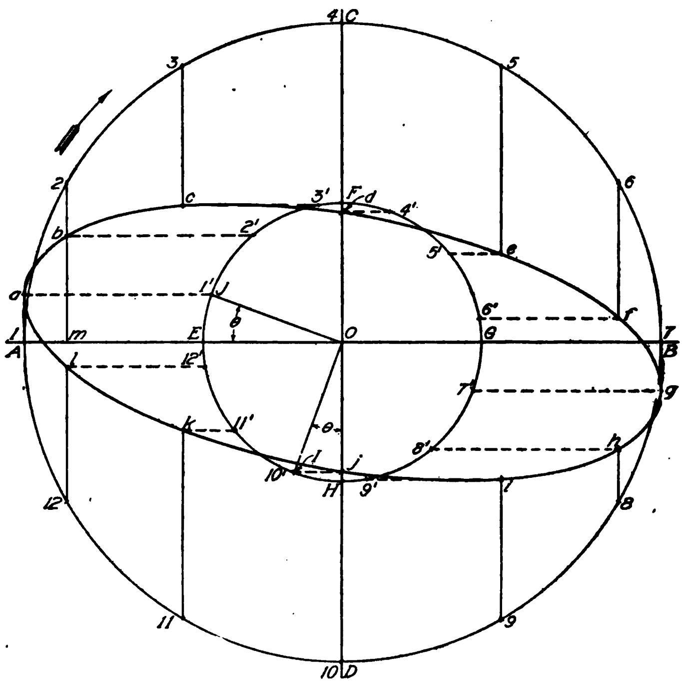


Fig. 57. Valve Ellipse Diagram for Studying Valve Action

may be obtained. Having the valve travel circle established, all the events of the stroke may be found as has already been pointed out in the study of the Zeuner diagram.

Valve Ellipse Diagram. The valve ellipse diagram furnishes another method for studying the valve action, aside from that furnished by the Zeuner diagram. The valve ellipse has been used a number of years as a means for representing the relative positions of the valve and the piston.

The principle of its construction as applied to the arrangement of valve and rods, as shown in Fig. 55, is to draw lines at right angles to each other, one representing the travel of the piston, the other that of the valve. Thus, in Fig. 57, draw the circle $ABCD$, having a diameter equal to the stroke of the piston drawn to a predetermined scale. This circle represents the path of the crank pin center. Divide this circle into any number of equal divisions, in this case, twelve, at points *1*, *2*, *3*, etc. It is evident that if a line be drawn from any one of these points, as *2*, perpendicular to the line AB, that, neglecting the angularity of the connecting rod, the distance $A\,m$ would represent the displacement of the piston as the crank moved forward from A. To allow for the angularity of the connecting rod, take a radius equal to the length of the connecting rod drawn

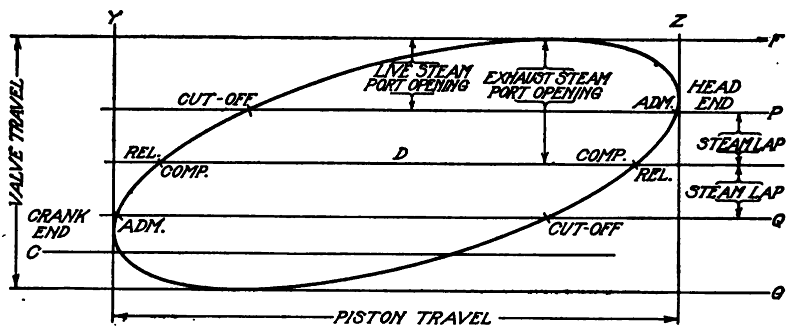


Fig. 58. Valve Ellipse Diagram Showing Information to be Obtained from its Analysis

to the same scale as that of the circle $A\ B\ C\ D$ and sweep the arcs from the points *1*, *2*, *3*, etc., with the center on the line $A\ B$ produced. Now representing the path of motion of the valve by the line $H\ F$, drawn perpendicular to $A\ B$, the eccentric position $O\ I$—which is located at the angle $(90-\theta)$ degrees behind the crank—is, for the sake of convenience in constructing the ellipse, located at $O\ J$. Having drawn the valve travel circle $E\ F\ G\ H$, begin at J and lay it off into the same number of equal parts as was done in the case of the crank circle. For the crank position A, the corresponding eccentric position is J, and hence, by projecting a vertical line and a horizontal line from A and J, respectively, the point a is located. In the same manner, the points b, c, d, etc., are located, thus completing the construction of the ellipse. The ellipse may have

different inclinations to the reference line, depending on conditions. This difference will be noted in comparing Figs. 57 and 58.

Thus far the discussion has dealt only with the construction of the ellipse. It is now proposed to point out what information may be obtained from the valve ellipse and for the sake of clearness, another figure is shown. After constructing the ellipse or having obtained it directly by an instrument specially constructed for the purpose, draw the reference line *C* in Fig. 58. Tangent to the ellipse, draw the lines *F* and *G* parallel to *C*. The distance between the lines *F* and *G* represents the travel of the valve. Midway between *F* and *G* draw the line *D*, the center line of the extreme travel of the valve. Assume that the valve is an ordinary plain slide valve having $1\frac{1}{8}$ inches steam lap and the zero exhaust lap, or line and line. Draw the lines *P* and *Q* $1\frac{1}{8}$ inches on each side of the center line *D*, and where these lines cut the ellipse determines the points where admission and cut-off occur for the two ends of the cylinder, as indicated in Fig. 58. Since there is no exhaust lap, the point where the line *D* cuts the ellipse gives compression and release for the two ends of the cylinder. In this case, the compression occurs on the head end at the same time that release occurs on the crank end, and *vice versa*. If the valve be given exhaust lap, it would be laid off in the same manner as the steam lap. Draw the lines *Y* and *Z* tangent to the ellipse and perpendicular to the reference line *C*. The distance between these two lines represents the length of the stroke of the engine drawn to scale. To find the per cent of the stroke at which any event occurs, it would be only necessary to drop a perpendicular to the center line *D* from the point on the ellipse corresponding to the event under consideration and obtain the percentage as previously explained. If the width of the steam port be known, it would be laid off from the lines *P* and *Q* toward the lines *F* and *G* as indicated. Assuming admission on the head end to occur as marked on the line *P*, it is evident from the portion of the curve contained between the lines *F* and *P* that at the beginning of the stroke the steam port was opened rather quickly and that cut-off occurred by the port being closed very slowly. During this time, the piston moved approximately three-quarters of the stroke. There being no exhaust lap or inside clearance, release occurred when the valve reached its central position. At the same

time that head-end release took place, compression began on the crank end, then followed crank-end admission, cut-off, release, and head-end compression, in regular order.

The valve ellipse has been largely used by steam railroad engineers and, as a result of the demand for such information as can be obtained from a consistent study of it, several devices have been invented for taking the ellipse directly from the engine. These devices consist of a drum the circumference of which is made proportional to the stroke of the engine. A sheet of paper is held on this drum by means of clips somewhat in the same manner as are found on the drums of steam engine indicators. This drum is mounted on a frame and when in use is placed in a convenient position above the crosshead or on the steam chest in such a position that its axis of rotation is perpendicular to the direction of the motion of the valve. The drum is rotated by means of a cord connection with the crosshead. Attached to the apparatus is a pencil which receives the same motion as that of the valve by means of a connection with the valve rod. Hence by the combination of the two movements, that is, of the drum moving with the piston and that of the pencil moving with the valve, the valve ellipse diagram is drawn.

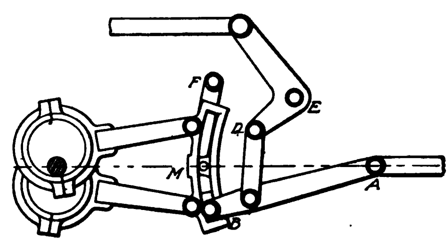


Fig. 59. Gooch Link Motion

From the study of the Stephenson gear, it is obvious that it is very flexible, and that it is readily adjusted to all irregularities of operation. Great care must be taken in its design in order that it may properly perform its work. Owing to the large number of parts and the size of same on large engines, it frequently gets out of alignment, its parts wear considerably and, on locomotives, the lubrication is sometimes difficult. On this account, it requires frequent attention in order that the best results may be obtained. All things considered, it is doubtful whether any other reversing gear gives as good a steam distribution as does the Stephenson gear when it is properly adjusted and operated.

Gooch Link. The Gooch link, illustrated in Fig. 59, has been extensively used on European locomotives, although it is gradually

being replaced by a type of valve gear known as the Walschaert, which will be described later.

The Gooch link has its concave side turned toward the valve instead of toward the eccentric. The radius of curvature of the link is equal to $A\ B$, the length of the radius rod. The link is stationary and the link block slides in the link. The engine is reversed by means of the bell-crank lever on the reverse shaft E which shifts the link block instead of the link, as is the case with the Stephenson. The link is suspended from its saddle pin M, which is connected by a rod to the fixed center F so that the link can move forward and back as the eccentricity is changed, or it can pivot about its saddle pin as the eccentrics revolve.

Since the radius of the link arc is equal to $A\ B$, it is apparent that the block can be moved from one end of the link to the other, that is, from full gear "forward" to full gear "back" without moving the point A, which is on the end of the valve rod. The lead then is constant for all positions of the block. The gear is more complicated than the Stephenson and requires nearly double the distance between shaft and valve stem.

RADIAL TYPE OF VALVE GEAR

In general, it would be desirable to have precisely similar steam distribution at each end of the cylinder, and it would often be of great advantage with a gear like the Stephenson if the cut-off could be shortened without changing any other event of the stroke. A Stephenson gear can be made to maintain equality of lead for both ends of the cylinder as the cut-off is shortened, but we have seen that in so doing, the lead of both ends is either increased or diminished according as the link is arranged with "open rods" or "crossed rods." Moreover, the compression is hastened by bringing the link to mid gear, all of which in many instances is undesirable.

This disadvantage of the Stephenson link motion led to the design of the so-called "radial valve gears," many of which are so complicated as to be impracticable, but all of which obtain a fairly uniform distribution of steam.

Hackworth Gear. The essential features of the Hackworth gear are indicated in outline in Fig. 60. In this figure S is the center of the shaft, and the eccentric E is set 180 degrees from the crank

SH. At the right-hand end of the eccentric rod EA is pivoted a block which slides in a straight, slotted guide. The guide remains stationary while the engine is running, but can be turned on its axis P to reverse the engine or to change the cut-off. P is a pivot which is located on a horizontal line through S in such a position that $D\,P$ is equal to $E\,A$. If these two distances are equal, A will coincide with P when the crank is at either dead point and the slotted guide may be turned from "full gear forward" (as shown in the figure)

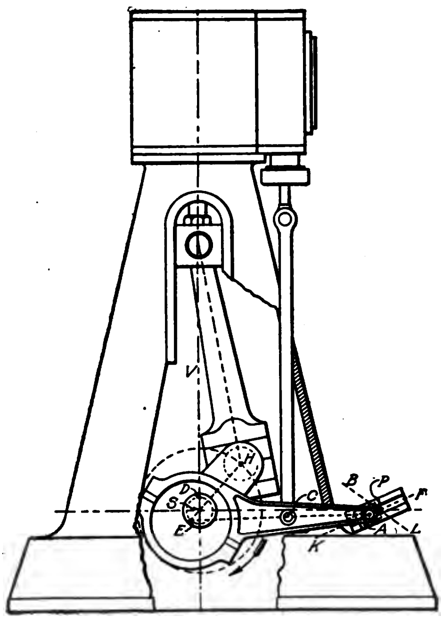


Fig. 60. Diagram of Hackworth Radial Valve Gear

through the horizontal position to "full gear back" (as shown by the line BL) without moving the valve. It will be observed, therefore, that the leads are constant for all positions of the guide. The valve rod running upward from C connects with the valve stem, which it moves in a straight line. The valve stem is made just long enough to equalize both leads and, if the point C has been properly chosen, the two cut-offs will be very nearly equal for all grades of the gear.

A somewhat better valve action is obtained by slightly curving the slotted guide, with its convex side downward. This gear is sometimes used on marine engines and on small stationary engines.

Marshall Gear. The most objectionable feature of the Hackworth gear is, perhaps, the slotted guide, for the sliding of the block causes considerable friction and wear. The Marshall gear, shown in outline in Fig. 61, is designed to eliminate this feature. The point A moves in the desired path by swinging on the rod FA about F as a center. While the engine is running, the lever $F P$ remains stationary, but can be turned on its axis P to reverse the engine or to change the point of cut-off. The pivot P is located precisely as

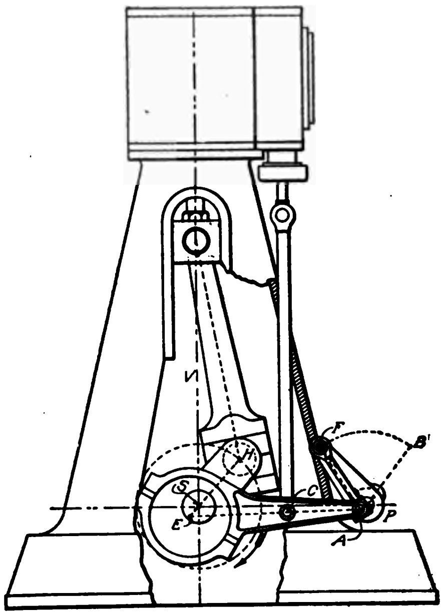


Fig. 61. Diagram of Marshall Radial Valve Gear

in the Hackworth gear, and the lever $F P$ can be turned from "full gear forward" (as shown in the figure) to "full gear back" (as shown by the line BP), intermediate positions giving different cut-offs the same as with the Hackworth gear. Since $F A$ is made equal to $F P$, the point A will always swing through P no matter where F may be and will coincide with P when the engine is on dead center. The leads for all positions of the gear, therefore, will remain constant, as in the preceding case.

The Marshall gear is sometimes made with the point C located on the right of A, on the line $E A$ produced. In this case, if the same kind of valve is to be used, the eccentric E must move with

the crank instead of 180 degrees from it. The Marshall gear is frequently used on marine engines, the one eccentric being simpler than the two required by the Stephenson.

Joy Gear. The Joy radial gear, Fig. 62, is perhaps the most widely known, and is certainly one of the best radial gears. It is frequently used on marine engines and on some English locomotives. No eccentrics are used, the valve motion being taken from *C*, a point on the connecting rod. *H* is a fixed pivot supported on the cylinder casting. The lever *ED* has a block pivoted at *A*, which slides back and forth in a slotted guide, having a slight curvature, the concave side being toward the right. The guide and the lever *PF* are fastened to the reverse shaft *P* and, by means of a reverse rod leading off from *F*, can be turned from full gear forward, as shown, to full gear back, when the pin *F* moves over to the position *B*. Motion is transmitted to the valve stem by means of the radius rod *EG*.

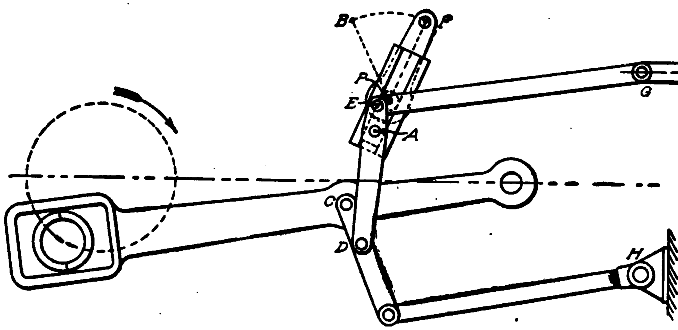


Fig. 62. Diagram of Joy Radial Valve Gear

The proportions are such that when the crank is on either dead point, the pivot of block *A* coincides with *P*, so that the curved guide may then be set in any position without moving the valve; therefore the leads are constant. This gear gives a rapid motion to the valve when opening and closing and a more nearly constant compression than the Stephenson gear, and the cut-off can be made very nearly equal for all positions of the gear. Its many joints cause wear and its position near the crosshead makes a careful inspection of the crosshead and piston rather difficult while the engine is running.

Walschaert Gear. The Walschaert gear, Fig. 63, stands today as the best representative of the radial gear type. It has for many

years been very largely used on all the important European railroads. It has been used considerably in England and at the present time is being applied to a great many locomotives in America.

Analysis of Valve Motion. When the Walschaert gear is used, the valve receives its motion from two distinct sources, namely, from the crosshead and from the eccentric crank. In Fig. 64 the various parts of the gear are named. The crosshead connection gives to the valve a movement equal to the lap plus the lead, at the extremities of the stroke, when the eccentric crank is in its mid-position. The eccentric crank leads the main crank by an angle of 90 degrees for a valve having external admission, and follows the

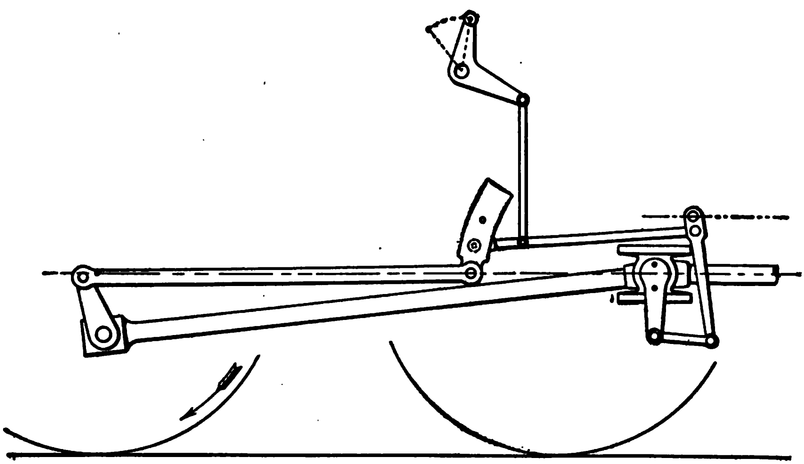


Fig. 63. Diagram of Walschaert Radial Valve Gear

main crank by 90 degrees for an internal admission valve. Locating the eccentric crank exactly 90 degrees in advance or behind the main crank is one of the necessary adjustments of the Walschaert gear. It is evident, therefore, that if an eccentric rod of proper length be attached to the eccentric crank and the valve through proper means, when the engine is on dead center, the valve would be in mid-position. However, in order to have economic operation, it becomes necessary to have some lead at dead center positions, hence the valve must be displaced by an amount equal to the lap plus the lead. Since the eccentric crank must be 90 degrees from the main crank, some other means must be used to

obtain the proper displacement and the method of accomplishing this on the Walschaert gear is one of its most distinguishing features. An attachment is made between the crosshead and the valve stem by means of a lever known as the combination lever, or, as shown in Fig. 64, the lap and lead lever.

In order to obtain the proper displacement of the valve when the engine is on dead center, the attachment of the combination lever to the crosshead and to the valve rod must bear a definite ratio to the stroke and valve travel. In other words

$$S:t \text{ as } L:V$$

or

$$V=\frac{Lt}{S}$$

in which S is stroke of piston in inches; t is twice the sum of the lap plus the lead in inches; L is distance between the crosshead connec-

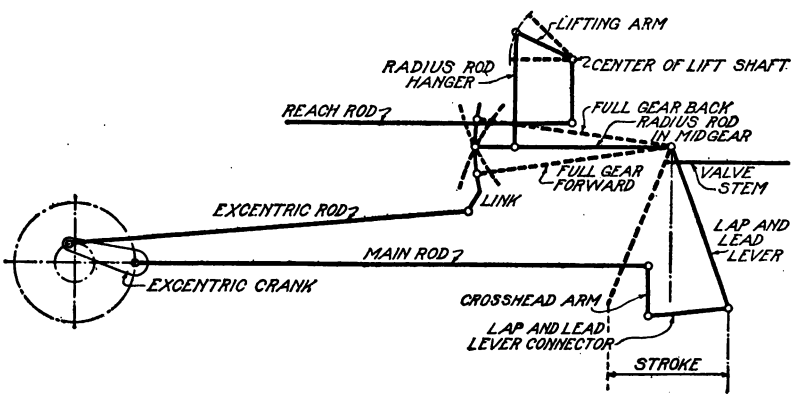


Fig. 64. Diagrammatic Analysis of Walschaert Valve Gear Action

tion and that of the radius lever in inches; and V is distance between the connection of the radius lever and that of the valve stem in inches. The above expression holds good for either an inside or an outside admission valve.

When using an inside admission valve, the connection between the radius rod and the combination lever is made above the valve stem connection as shown in Fig. 64, that shown in Fig. 63 being the arrangement for an external admission valve.

Link Motion. We have thus far traced the movement of the valve, taking into consideration the crosshead and eccentric crank connection and omitting for the sake of clearness the link connection. It will be noted that the link is pivoted at the center. The link block is raised or lowered by means of the reverse lever and bell crank. The link block is connected to the radius rod, which has a length equal to that of the link; hence, when the engine is on either dead center, the link block can be raised from one extreme position to the other without moving the valve. Therefore this gear, if properly constructed, gives a constant lead for all positions of the reverse lever. The proper construction, suspension, and attachment of the link to its allied parts is a very important matter and one rather difficult to accomplish. The proper location of the attachment of the link to the eccentric rod gives the designer a great deal of trouble, in obtaining the desired action of the valve. In locating the longitudinal position of the link fulcrum, consideration must be given to the length of the eccentric rod, which should have a minimum length of three and one-half times the throw of the eccentric and should be made as long as the existing conditions will permit. It should be so located that the radius and eccentric rods are approximately of equal length. The point of connection between the link and the eccentric rod should be as near the center line of motion of the connecting rod as possible, making due allowance for the angularity of the rods. To accomplish this, it often happens that the throw would be excessive. In such cases, a compromise is necessary, the point of connection being raised above the center line of motion as the case demands. It has been found in designing this gear that these considerations require shifting the eccentric crank from one to two degrees, thus making the angle between the main crank and the eccentric crank 91 degrees or 92 degrees instead of 90 degrees, as theoretically required. The angle being increased by such a small amount does not affect the movement of the valve to any appreciable extent.

Adjustment of Gear. From the foregoing brief remarks, it is to be noted that in order to secure the best results, the design of the Walschaert gear requires very accurate work. No hard and fast rules can be laid down as how to secure the best design, for each case presents different problems. The best way to secure required results is to try out the design on a model.

The American Locomotive Company gives the following suggestions for adjusting the Walschaert valve gear:

(1) The motion must be adjusted with the crank on the dead center by lengthening or shortening the eccentric rod until the link takes such a position as to impart no motion to the valve when the link block is moved from its extreme forward to its extreme backward position. Before these changes in the eccentric are resorted to, the length of the valve stem should be examined as it may be of advantage to plane off or line under the foot of the link support which might correct the length of both rods, or at least only one of these should need be changed.

(2) The difference between the two positions of the valve on the forward and back centers is the lead and lap doubled and can not be changed except by changing the leverage of the combination lever.

(3) A given lead determines the lap or a given lap determines the lead, and it must be divided for both ends as desired by lengthening or shortening the valve spindle.

(4) Within certain limits, this adjustment may be made by shortening or lengthening the radius bar but it is desirable to keep the length of this bar equal to the radius of the link in order to meet the requirements of the first condition.

(5) The lead may be increased by reducing the lap, and the cut-off point will then be slightly advanced. Increasing the lap introduces the opposite effect on the cut-off. With good judgment, these qualities may be varied to offset other irregularities inherent in transforming rotary into lineal motion.

(6) Slight variations may be made in the cut-off points as covered by the preceding paragaph but an independent adjustment can not be made except by shifting the location of the suspension point which is preferably determined by a model.

Zeuner Diagram for Walschaert Gear. The Walschaert gear may be examined by the aid of a Zeuner diagram to the same limited extent as the Stephenson. The construction of the Zeuner for a Walschaert gear is somewhat easier than for the Stephenson because the locus of the virtual eccentric centers lie on a straight line, due to the constant lead. For example, take a maximum valve travel of $5\frac{1}{2}$ inches, a lap of 1 inch, and a lead of $\frac{3}{16}$ inch. In Fig. 65 (scale of drawing is exactly three-fourths size), the valve travel circle is $ABCD$, the lap circle $ILMN$. Lay off the given lead IF $\frac{3}{16}$ inch and through the point F erect a perpendicular line cutting the circle $ABCD$ at H and E, thus locating the two eccentric positions. Since there is a constant lead for any valve travel, the line HFE becomes the locus of the virtual eccentric centers. Assuming a cut-off of 80 per cent, locate the line OK and at J, the point where this line cuts the steam-lap circle, erect a perpendicular line

and extend it until it cuts the line *H F E* at *G*. The point *G* is the extremity of the valve travel circle for 80 per cent cut-off, the radius

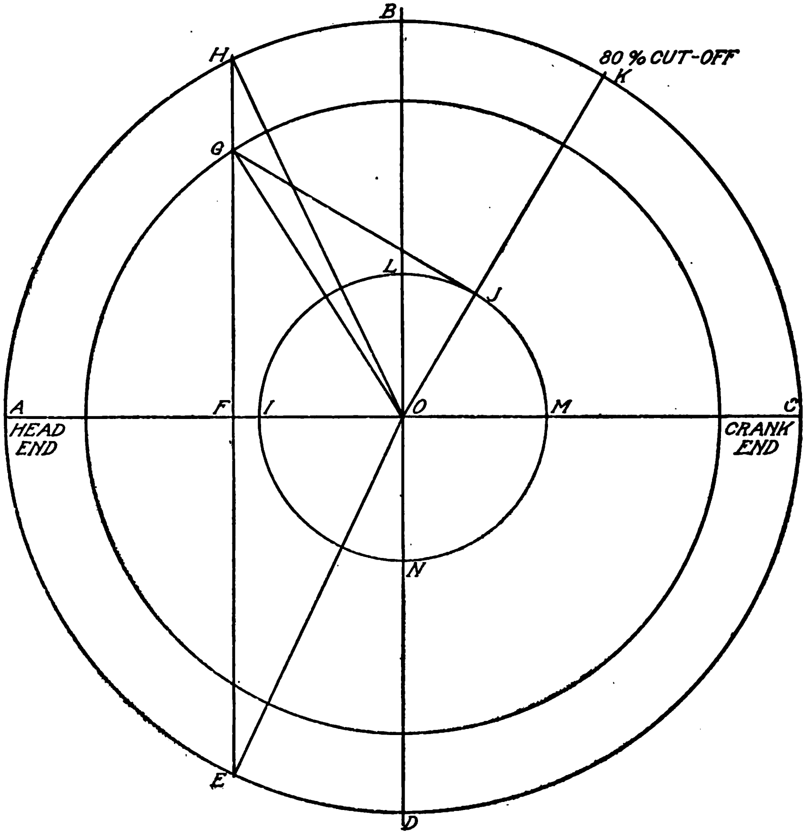


Fig. 65. Zeuner Diagram for Walschaert Gear

Fig. 66. Application of Walschaert Valve Gear to a Passenger Locomotive

of which will be *O G*, $2\frac{3}{16}$ inches. In like manner, the valve travel can be obtained for any other point of cut-off.

Dimensions of Walschaert Gear Parts. For an engine such as is shown in Fig. 66, an approximate value of the various rods and levers may be taken as follows. By referring to Fig. 64 the location of the various parts can be determined more readily than in Fig. 66.

Main rod	8′-0″	Radius rod	3′-10″
Eccentric crank	1′-2″	Lap and lead lever (total)	3′- 0″
Eccentricity	6½″	Lap and lead lever connector	1′- 2″
Eccentric rod	4′-6″	Crosshead arm	1′- 0″
Link arc	1′-10″	Stroke	2′- 0″

DOUBLE VALVE GEARS

It has been shown that a plain slide valve under the control of a gear that gives a variable cut-off, such as a shifting eccentric or a link motion, will not give a satisfactory distribution of steam at a short cut-off owing to excessive compression, variable lead, or early release. These difficulties are overcome in a measure by the use of the radial gear; and also by the use of two valves instead of one.

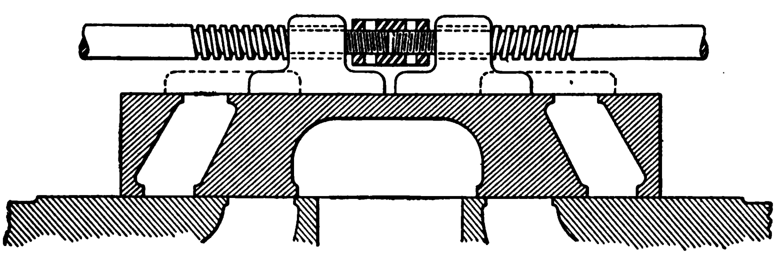


Fig. 67. Section of Meyer Double Valve

The main valve controls admission, release, and compression; the other, called the cut-off valve, regulates the cut-off only, which may be changed without in any way affecting the other events of the stroke. This cut-off valve, sometimes known as the riding cut-off valve, may be placed in a separate valve chest, or it may be placed on the back of the main valve.

Meyer Valve. The most common form of double valve gear is the Meyer valve, Fig. 67. The cut-off valve is made in two parts and works on the back of the main valve. The two parts are connected to a valve spindle with a right-hand and a left-hand thread, so that their positions may be altered by rotating the valve spindle.

A swivel joint is usually fitted in the valve spindle between the

steam chest and the head of the valve rod, and the valve spindle is prolonged into a tail rod which projects through a stuffing box on the head of the steam chest, Fig. 68. The end of this tail rod is square in section and reciprocates through a small hand wheel, by means of which it can be rotated while the engine is running, whatever the position of the valve may be.

Each valve is under the control of a separate eccentric. The eccentric which moves the main valve is usually fixed, while the cut-off valve eccentric may be under the control of a governor. Since a slight compression is desired, the main valve is set to give late cut-off and this will also give late release and late compression, and allow a wide range of cut-off for the cut-off valve. With this gear, lead, release, and compression are entirely independent of the ratio of expansion, and the cut-off is much sharper, because the cut-off valve, when closing the ports, is always moving in a direction opposite to that of the main valve. The valve may be designed by means of the Zeuner diagram.

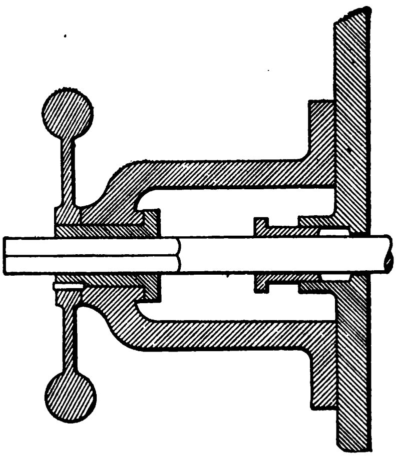


Fig. 68. Stuffing Box and Valve Spindle of Meyer Gear

Design by Zeuner Diagram. Let us design a Meyer valve having an eccentricity of 2 inches. Let the eccentricity of the cut-off valve be $2\frac{1}{4}$ inches and the relative travel of the cut-off valve in relation to the main valve be 3 inches. This will make the relative motion of the cut-off valve equivalent to the travel of a plain slide valve with an eccentricity of $1\frac{1}{2}$ inches. Let the outside lap on the main valve be $\frac{3}{4}$ inch, the lead $\frac{1}{32}$ inch, the compression 5 per cent of the stroke, and let the ratio of the length of the crank to connecting rod be 1 to 6.

In Fig. 69, draw $X\ O\ Y$, the main valve travel, equal to 4 inches. Lay off $X\ D$ equal to 5 per cent of 4 or 0.2 inches, and with a radius of 12 inches, and the center on $Y\ X$ produced, draw the arc $D\ H\ K$. $H\ K\ O$ is the crank position at compression on the head end; $C\ K\ O$, the crank position at compression on the crank end, is found in a

similar manner. Lay off $O\ I$ equal to the lap plus the lead, and draw the valve circle for the main valve through I and O with a diameter equal to its eccentricity of 2 inches. To do this, take a radius equal to 1 inch, and draw arcs from I and O as centers that shall intersect at B. B is the center of the valve circle and $O\ B\ E$ is the eccentricity, 2 inches. With E as a center, and with a radius equal to half the relative travel of the cut-off valve (in this $1\frac{1}{2}$ inches) draw an arc.

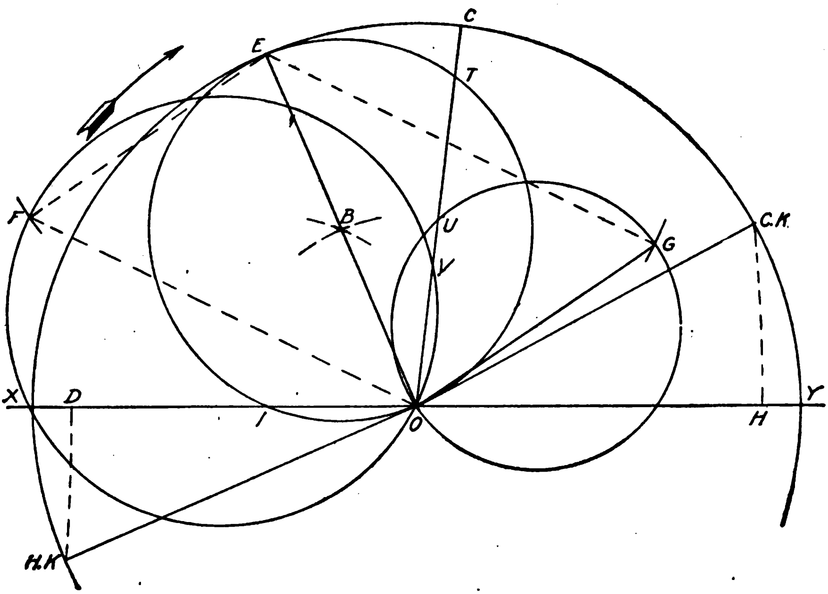


Fig. 69. Zeuner Diagram for Meyer Valve Gear

With O as a center and with a radius equal to $2\frac{1}{4}$ inches, the eccentricity of the cut-off valve, draw another arc intersecting the first one at F. On $O\ F$ as a diameter, construct a valve circle. This valve circle will represent the absolute motion of the cut-off valve, independent of the motion of the main valve. This circle then will show the displacements of the cut-off valve from the center of the steam chest. With E as a center and with a radius equal to $F\ O$, draw an arc, and with O as a center and with a radius equal to $E\ F$, draw another arc intersecting the first at G. On $O\ G$ as a diameter, construct a valve circle. This circle will then represent the travel of the cut-off valve moving on the main valve. That is, it will represent the

displacements of the cut-off valve from the center of the main valve. This circle is not, properly speaking, a valve circle, and $O\,G$ is not an eccentricity, but simply represents the relative motion of the two valves. This can be proved by analytical geometry, but an inspection of the figure shows that this must be true.

Draw the crank line $O\,C$ at any position, cutting the valve circles at T and U and V. $O\,V$ represents the absolute displacement of the cut-off valve, that is, from the center of the steam chest, and $O\,T$ represents the displacement of the main valve. The relative displacement of the cut-off valve, that is, from the center of the main valve, will be the difference between $O\,V$ and $O\,T$, since both valves are moving in the same direction. By careful measurement it will be found that $O\,U = O\,T - O\,V$, and any arc as $O\,U$ on the auxiliary circle $O\,U\,G$ will correctly represent the displacement of the cut-off valve from the center of the main valve at the corresponding crank angle.

In Fig. 70 are shown $H\,K$ the crank angle at head-end compression, $C\,K$ the crank angle at crank-end compression, the main valve circle, and the auxiliary circle, all of which have been transferred from Fig. 69. In order to avoid confusion, the construction lines and all lines not essential to the figure are omitted.

Lay off on Fig. 70, $O\,I$ equal to the outside lap $\frac{3}{4}$ inch and draw the head-end lap circle $H\,E\,O$. It will intersect the valve circle for the main valve at L and M. Draw $H\,O$ through L, representing the crank position at admission (head end) and $O\,M\,H$ through M showing the crank position at cut-off. This gives the greatest possible cut-off. The cut-off valve may be set to give a much earlier cut-off than this, but of course, a later setting would be of no avail, for the port would be closed by the main valve at this angle. The crank line $O\,M\,H$ cuts the auxiliary circle at N_1 so that $O\,N_1$ ($1\frac{15}{32}$ inches) is the clearance of the cut-off valve. That is, the edge of the cut-off valve must be set $1\frac{15}{32}$ inches from the edge of the main valve port in order to cut off at this crank angle. The full lines of Fig. 66 show the cut-off valve placed in this position.

The intersection of $H\,K\,O$ with the lower valve circle gives the inside lap at the head end of the cylinder. This line comes so nearly tangent to the valve circle that the intersection can be determined only by dropping a perpendicular to $H\,K\,O$ from E_2. This cuts the

circle at *P*, and *O P* equals the head-end inside lap $\frac{1}{32}$ inch, and *H E I* represents the corresponding lap circle.

The crank position at compression on the crank end is *C K*, which cuts the upper valve circle at N_2, giving an inside lap for the

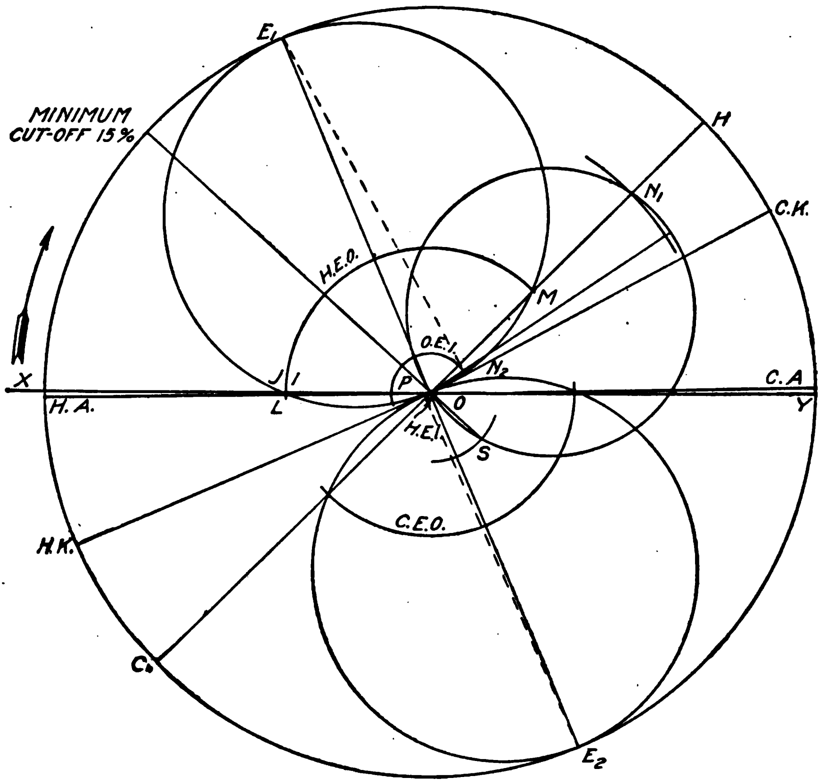


Fig. 70. Zeuner Diagram Showing Additional Factors in Design of Meyer Valve

crank end, of $O\ N_2$, $\frac{13}{64}$ inch. To make this intersection more apparent, the perpendicular may be drawn from E_1, as previously explained.

Suppose that it was required that the minimum cut-off should be 15 per cent. Find the crank position at 15 per cent of the stroke in the same manner as the crank position was found at compression. Produce this line through *O* until it cuts the auxiliary circle at *S*. Then *O S* equals the required lap $\frac{23}{64}$ inch for the cut-off valve in order to cut off at 15 per cent of the stroke. The dotted lines in Fig. 67 show the cut-off valve drawn in this position.

For a valve of this sort, the cylinder port should be $1\frac{1}{2}$ inches wide and the valve port 1 inch wide. Fig. 67 shows this valve laid out to scale, but as this process is in all respects similar to that described for laying out a plain slide valve, it will not be described in detail.

Shifting Eccentric Valve Gear. In addition to the valve gears already described, there is another class which receives a very large application, particularly in small and medium-power high-speed engines. This class of gear is what may be termed the shifting

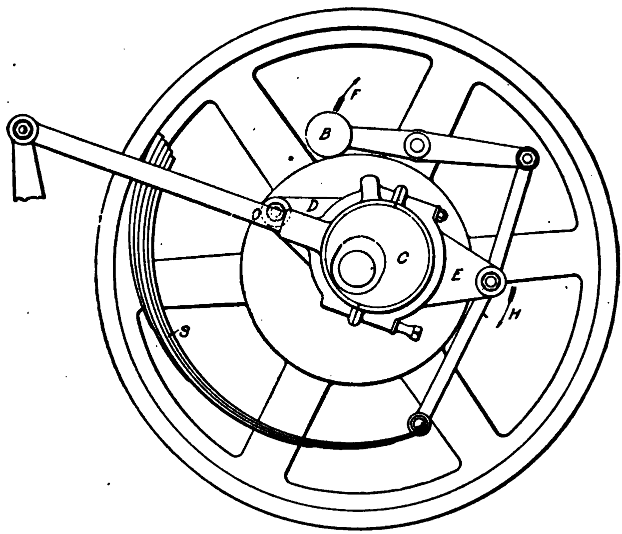


Fig. 71. Diagram Showing Action of Straight-Line Type of Shaft Governor

eccentric gear. The valve itself is the ordinary flat slide or piston valve, and the valve stem, eccentric rod, and eccentric are the same as used in the common arrangement. The difference between the fixed and the shifting eccentric lies in the method of attaching the latter to the shaft and in the mechanism provided to move this eccentric from one position to another across the shaft. The general arrangement is illustrated in Fig. 71, which represents that used on the straight-line engine.

In this, O is the fixed pivot of the eccentric lever E, and C is the eccentric. The pin H of the eccentric lever is connected through a link to a leaf spring and through the other to a weighted lever B, as shown. When the engine is running, the position of the weight B

changes under different speeds and loads, and this change in position is transmitted to the eccentric. Since O is a fixed pivot, any motion of the eccentric lever E or eccentric C must be around O as a center. Consequently, when the eccentric position changes, its center will move in a path which is an arc of a circle with a center at O. The slot

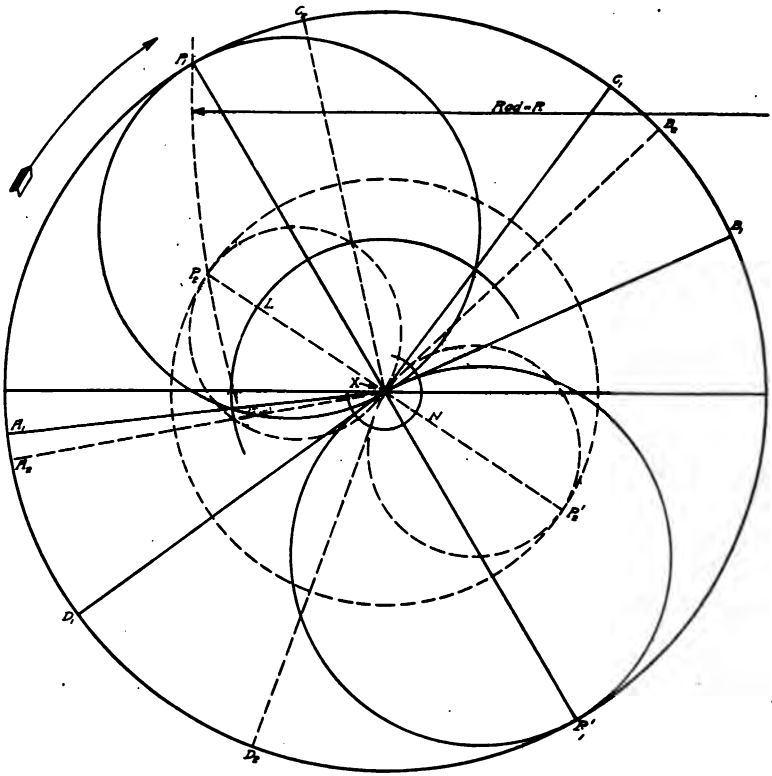


Fig. 72. Zeuner Diagram for Shifting Eccentric Valve Gear

in the eccentric is provided so as to enable it to move across the shaft to whatever position may be desired.

Two things are to be noticed for different eccentric positions. The first is that the eccentricity becomes less as the eccentric center moves along its path toward the shaft. The second is that the angular advance increases under this same condition. The first effect is to decrease the valve travel and the second to increase the angular

advance. The effect of the shifting of the eccentric on the motion of the valve is a combination of these two changes.

Fig. 30 shows the effect of changing the angular advance, and Fig. 31 shows the effect of changing the valve travel. The effect of both these changes acting together is shown in Fig. 72, which is a Zeuner diagram for a shifting eccentric valve gear.

Analysis of Zeuner Diagram. In Fig. 72 the full lines represent eccentric and crank positions at the point of maximum cut-off, and the dotted lines represent their positions corresponding to some earlier cut-off. A_1 is the crank position at admission; C_1 is its position at cut-off; B_1 is its position at release; and D_1 is its position at compression. P_1 and P_2 are the corresponding positions of the eccentric center, the subscripts referring to the maximum cut-off and to the earlier cut-off, respectively. The radius R is used to draw the path of the eccentric center and has the same length as the distance from the fixed eccentric lever pin to the center of the eccentric. The steam lap in the figure is XL and the exhaust lap is XN. The construction is made for the head end only, to avoid confusion due to the large number of additional lines required for that of the crank end. These laps remain the same for all positions of the eccentric.

Each of these two diagrams is made in exactly the same way as the ordinary Zeuner diagram and, if given the necessary data, the construction of the combined diagram should give no trouble. One point to bear in mind in drawing a Zeuner for this kind of valve gear is that the eccentric center for any position of the gear will lie somewhere along the arc described with the radius R.

An inspection of Fig. 72 shows that all events occur earlier with the earlier cut-off. However, they do not all continue for the same period, as was found to be the case when the angular advance alone was changed. This is because the valve travel is changed with the angular advance. The combined effect is that admission is advanced very slightly, cut-off is advanced a considerable amount, and release and compression are each advanced a moderate amount. Since the cut-off advances a greater amount than the release, the result is a greater expansion at earlier cut-off, which is more economical, within reasonable limits, in the use of steam. Also the increase in compression up to a certain point will make the engine run more smoothly by cushioning the piston better on the dead centers. Another effect of

earlier cut-off is a decrease of lead from that at maximum cut-off. This may or may not be an advantage, depending on the engine speed, construction of steam ports, amount of clearance, etc.

Thompson Automatic Valve Gear. The Thompson automatic valve gear, commonly known as the "Buckeye", belongs to the general class of double valve gears. Its principle of action involves certain ingenious points which make its study very interesting.

Two styles of valves of this type have been developed by the Buckeye Engine Company, namely, a flat valve and a round or piston

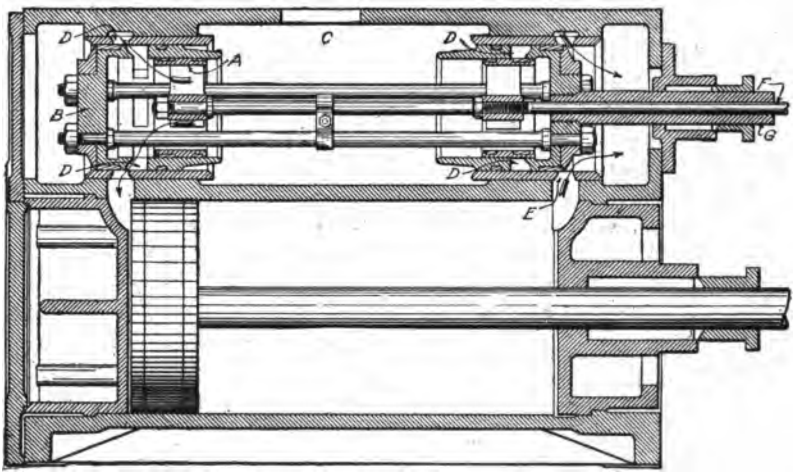


Fig. 73. Section of Cylinder and Valve-Gear Mechanism of Thompson Automatic Valve Gear

valve. While the essential features of the two valves are the same, the piston valve, illustrated in Fig. 73, is the simpler of the two and represents the latest practice. The cut-off valve *A* moves inside of the main valve *B*, and live steam entering at *C* passes through the cut-off ports *D D* in the main valve and is admitted to the cylinder, as these ports are alternately brought into coincidence with the cylinder ports *E E*, as shown by the arrows on the left. The exhaust steam is discharged at the ends of the main valve and does not come in contact with the valve except at the ends. It will be noted that the construction is such that the valve is at all times balanced.

The valve stem *F* of the cut-off valve passes through the hollow stem *G* of the main valve. Packing rings are used on both valves

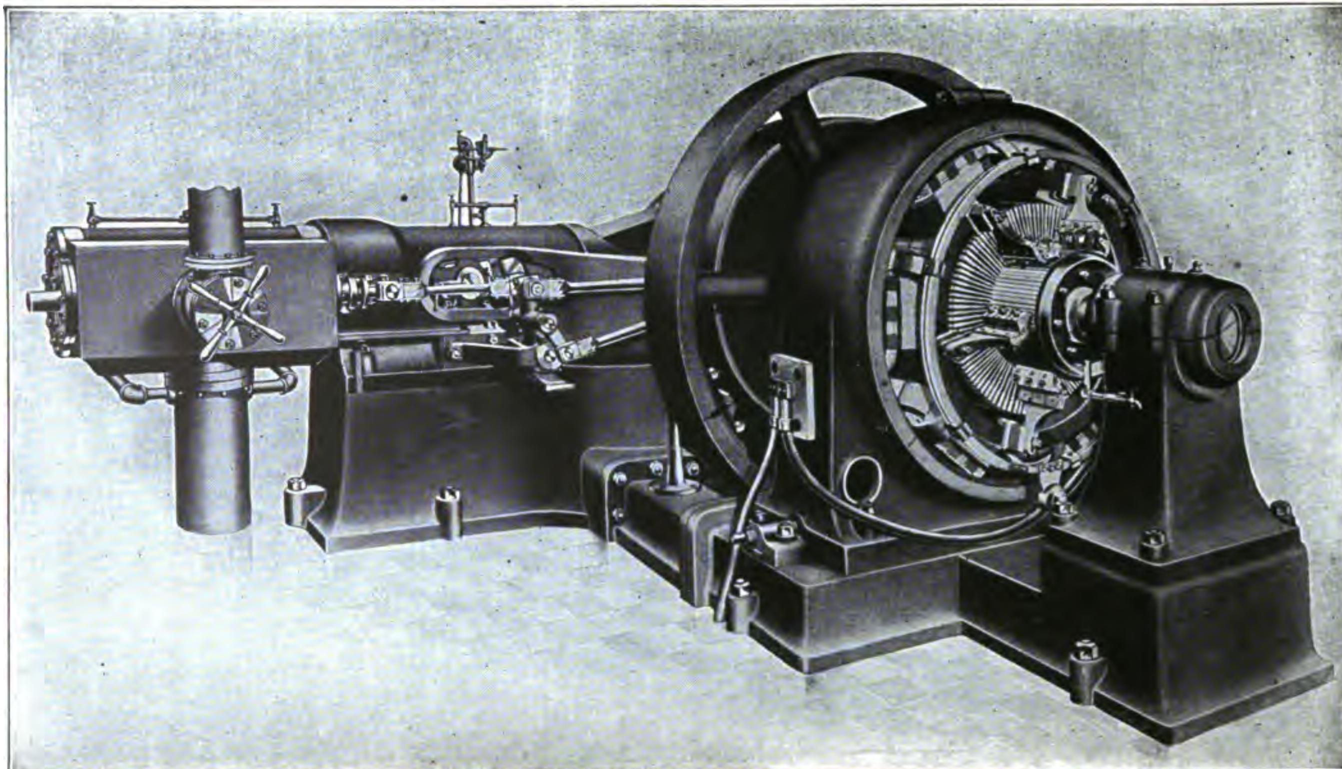


Fig. 74. Thompson Automatic Valve Gear Applied to Buckeye Engine
Buckeye Engine Company, Salem, Ohio

to insure a steam-tight connection, and bridges are provided to afford a proper bearing for the rings in passing over the ports.

Fig. 74 shows the valve gear as applied to the engine. The operation of the gear can be better understood by referring to the line drawing, Fig. 75. The crank $A\ C$ is shown on the head-end dead center and running over. The eccentric D connects to the main valve stem through the eccentric rod $D\ M$, the joint M being guided by the rocker arm $H\ M$, pivoted to the engine frame at H. The cut-off valve is operated from the eccentric E by the eccentric rod $E\ F$, and the rocker arm $F\ K\ N$ is pivoted to the rocker arm $M\ H$ at

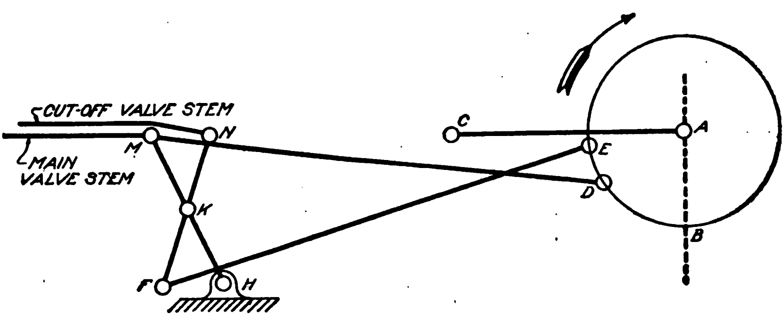


Fig. 75. Diagram Showing Operation of Thompson Automatic Valve Gear

the point K. In the compound rocker arm, the arms $M\ K$, $K\ H$, $N\ K$, and $K\ F$ are all equal. On account of the valve under discussion being one having internal admission, it will be noted that the eccentric of the main valve follows the crank instead of preceding it, as is found in most cases.

Starting from the head-end dead center, suppose the crank, and likewise the eccentric, to turn through a small angle ϕ. This movement will cause the point M and the main valve to move to the left, a distance which we will call x, and the pivot point K will also move to the left a distance $\frac{x}{2}$. Now if F be considered as a fixed point, the movement of K, equal to $\frac{x}{2}$, causes the point N, and consequently the cut-off valve, to move to the left a distance x. The point F is not fixed, for while M is moving a distance x to the left, the rocker arm $F\ K\ N$, being pivoted to the rocker arm $H\ K\ M$ at the point K, will cause the point F to move to the left (due to the rotation of the eccen-

tric E) a distance which we will call y. This movement of F would cause N to move to the right a distance equal to y, provided the point K were stationary. Thus it will be seen that the point N and the cut-off valve are given a movement which is the resultant of two motions and is equivalent to $x-y$; and the relative movement of the two valves would be $x-(x-y)=y$. But y is the motion which would be given the cut-off valve by the eccentric E, independent of the other mechanism; *i.e.*, the construction is such that the cut-off valve moves on the main valve in much the same manner as an

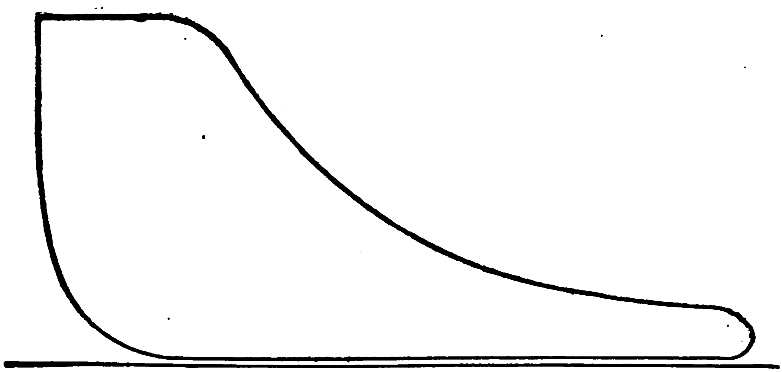


Fig. 76. Indicator Diagram, Showing Effect of Application of Thompson Gear

ordinary plain slide or piston valve moves over stationary ports when connected to a constant throw eccentric through a reversing rocker arm.

The governor in moving the cut-off eccentric simply causes it to turn about the shaft, thus cutting off the steam earlier or later, according as the eccentric is advanced or moved back on the shaft. This action takes place without changing the cut-off valve travel or the relative movement of the two valves, since the throw of the two eccentrics is equal and constant. The Zeuner diagram for the Buckeye valve is worked out in a manner similar to that described for the Meyer valve.

Two important claims are made for this valve gear:

(1) On account of the valves moving in opposite directions at the instant cut-off occurs, cut-off is made very quickly, thus eliminating quite largely wiredrawing and giving an indicator diagram having a sharp turn at the point of cut-off, resem-

bling that given by a Corliss valve gear. This is illustrated by the diagram, Fig. 76.

(2) On account of the constant travel of the valves, they wear better than those that control the regulation by varying the valve travel.

This latter claim makes the gear particularly suited for the piston valve, since uneven wear or leakage is more liable to result from the packing rings if the valve movements are variable.

DROP CUT-OFF GEARS

The ordinary slide valve controls eight different events of the stroke, that is, admission, cut-off, release, and compression for both ends of the cylinder. A change in the setting of a plain slide valve that affects any one event on the crank end, let us say, will also change to a greater or less degree every other event of the stroke, on the head end as well as on the crank end; so that in setting a slide valve, the desired position for one event must usually be sacrificed in order to make the others less objectionable.

In order to provide a better distribution of steam than is possible with a single valve, some engines have four valves, two at each end of the cylinder. In horizontal engines, two valves are placed above the center line of the cylinder and two below, the upper being for admission and cut-off, the lower for release and compression. Since each valve controls but two events, a very satisfactory adjustment can be made and the extra complication and cost for large engines are more than overbalanced by the advantages gained, viz, a very much better distribution of steam; short steam passages and small clearances; separate ports for the admission of hot steam, and the exhaust of the same steam after expansion when its temperature has fallen; and finally the possibility of opening and closing the ports very rapidly, thus preventing wiredrawing. The small clearances, short ports, and separate admission and exhaust materially reduce the cylinder condensation, and thus effect a large saving in the steam consumption.

When four valves are used for high speeds, the motions of all must be positive, that is, they must be connected directly to some mechanism that either pushes or pulls them through their entire motion, but for speeds up to 100 revolutions or so, a disengaging

mechanism may be used, and the valves may shut off themselves, either by virtue of their weight or by means of springs or dashpots.

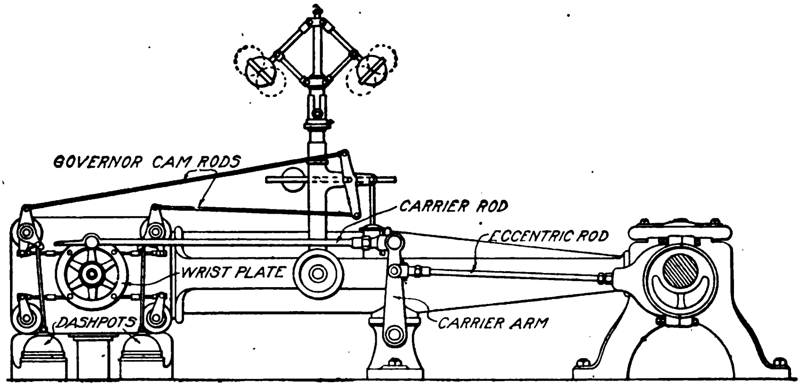


Fig. 77. Diagram of Reynolds-Corliss Drop Cut-Off Gear

The valve is usually opened by means of links or rods moved by an eccentric and, at the proper point of cut-off, the rod is disengaged from the valve, which drops shut, hence the term "drop cut-off" gears.

Reynolds-Corliss Gear. The most widely known drop cut-off gear is the Reynolds-Corliss, Figs. 77 and 78. It is often referred to as the *Reynolds hook-releasing gear.* An eccentric on the main shaft gives an oscillating motion to a circular disk, called the *wrist*

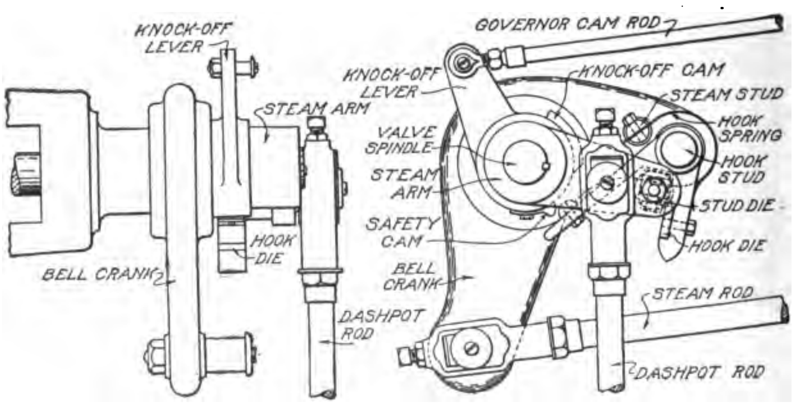


Fig. 78. Details of Reynolds Hook

plate, pivoted at the center of the cylinder. It transmits motion to each of the four valves through adjustable links known as *steam rods*

or *exhaust rods,* according to whether they move the admission or exhaust valves.

The valves which are shown in section in Fig. 79 oscillate on cylindrical seats, and the position of the rods is so determined that they give a rapid motion to the valve when opening or closing, and hold it nearly stationary when either opened or closed.

The Reynolds hook is shown in detail in Fig. 78. The steam arm is keyed to the valve spindle which passes loosely through a bracket on which the bell-crank lever turns, and the spindle is packed to make a steam-tight joint where it enters the cylinder. Motion of the steam rod toward the right will turn the bell-crank lever and raise the hook stud. The hook (from which the gear derives its name), pivoted on this stud, has at one end a hardened steel die with sharp, square edges, and at the other end, a small steel block with a rounded face. As the hook rises, the hook die engages the stud which is fastened to the steam arm, and one end of the steam arm is thus raised. This turns the valve in its seat and admits steam. As the hook continues to rise, its stud moves in an arc above the valve spindle, and the round-faced block at its left-hand end strikes the knock-off cam, which causes the hook to turn about its stud and disengage the hook die from the stud die. In raising the steam arm, the dashpot rod is also raised and a partial vacuum is created in the dashpot. As soon, therefore, as the dies become disengaged, the dashpot quickly drops under the force of this vacuum, thus turning the steam arm and closing the valve. The striking of the left-hand end of the hook against the knock-off cam determines the point of cut-off by releasing the valve at that instant.

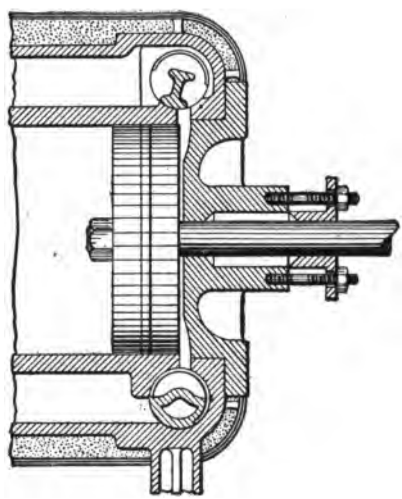


Fig. 79. Diagram, Showing Reynolds-Corliss Valves in Section

This cam is a part of the knock-off lever to which the governor cam rod is fastened. Any action of the governor which would cause

the cam rod to move toward the right would cause this knock-off lever to turn on its axis, the steam arm, and consequently lower the position of the knock-off cam. This would cause an earlier contact between the cam and the end of the hook, and consequently an earlier cut-off. By lengthening or shortening the governor cam rod, the point of cut-off can be adjusted to suit the engine load without changing the speed.

There is a limit to this adjustment, for it can be shown that a Corliss gear operated by a single eccentric can not be arranged to cut off later than half-stroke. Suppose the eccentric is set just 90 degrees ahead of the crank. Then it will reach its extreme position just as the piston gets to half-stroke. If, by that time, the hook which was rising and opening the admission valve has not yet struck the knock-off cam, it can not strike it at all, for any further motion will cause the hook to descend to its original position, that is, its position at the beginning of the stroke; the hook will not disengage from the steam arm stud at all and the bell crank will return, closing the valve in the same manner in which it opened it. Cut-off will then take place near the end of the stroke, but it will not be the sharp cut-off produced by the sudden drop when the dies are disengaged.

If the eccentric were set less than 90 degrees ahead of the crank, the cut-off could be arranged to occur later than half-stroke, but this is decidedly impracticable, for with such a position of the eccentric, the action of the valves at release and compression is spoiled. When it is necessary to cut off later than half-stroke, as sometimes happens on low-pressure cylinders of compound engines, it may be arranged for by means of two eccentrics, one set *more* than 90 degrees ahead of the crank to operate the exhaust valves, and one *less* than 90 degrees ahead to operate the admission valves.

The safety cam, Fig. 77, is an important part of a Corliss gear. If for any reason the engine governor should fail to act, due, for instance, to the breaking of its driving belt, the governor would drop to its lowest position, supply more steam to the engine, and allow it to run away. The safety cam prevents this by moving so far to the right that it strikes the hook when it descends to pick up the steam arm. The hook is consequently turned toward the right and then lifted without engaging the stud die; the valve consequently remains closed and the engine stops.

Nordberg Gear. The Nordberg type of drop gear is designed for high speed and hard service. Instead of having its steam arm supported on the valve spindle, it is supported on a bearing formed by an extension of the steam bonnet, and the arm is provided with two hook dies instead of one. To eliminate side strains these two dies are connected on either side of the dashpot rod. The release is accomplished by means of an extension of the steam arm which rides in a slotted cam. The position of the latter is under control of the governor. The dashpot is also carried by the steam bonnet and is located above the gear. The spring type of dashpot is used to secure positive action at high speed.

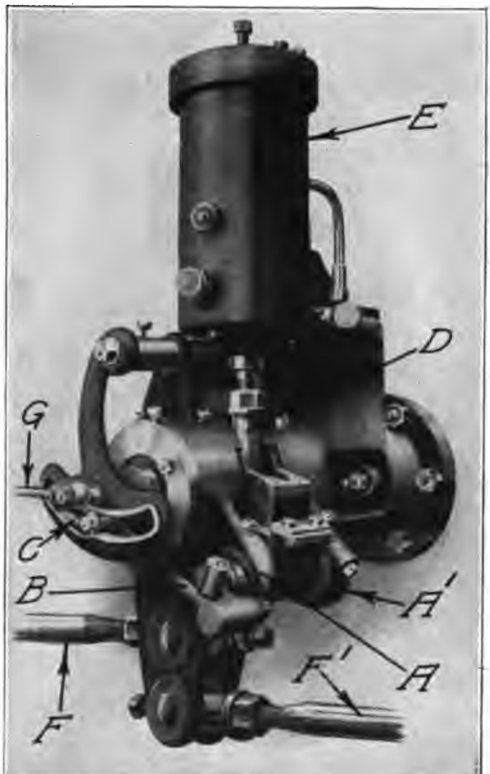


Fig. 80. Nordberg Drop Gear
Courtesy of Nordberg Manufacturing Company, Milwaukee, Wisconsin

In Fig. 80, *A A* are the two parts of the steam arm to which the hook dies (not shown) are attached. *B* is the extension of the steam arm, and it carries at its left end a roller which works in the releasing cam *C*. When this roller strikes the off-set, shown in the cam, it raises the arm *B*, which disengages the hook dies and allows the dashpot rod *D* to snap the steam valve shut. *E* is the dashpot cylinder. *F F* are the driving rods, the one on the right being connected to the eccentric on the engine shaft, and the one on the left driving the valve gear for the other end of the cylinder. The rod *G* is connected indirectly to the governor and controls the point of cut-off by changing its position as the governor changes, according to the load on the engine.

Brown Releasing Gear. In addition to the Reynolds hook, several other devices are in use for opening and releasing Corliss admission valves. Among them, the Brown releasing gear, Fig. 81, may be noted. The steam rod and dashpot rod are arranged much the same as in the Reynolds gear. The governor cam rod operates a plate cam having a curved slot so shaped that it takes the place of both the knock-off and the safety cam of Fig. 78. The steam arm is keyed to the valve spindle and carries at its lower end a steel die which is free to slip up and down a small amount. The part of this gear corresponding to the Reynolds bell crank becomes a straight rocker pivoted at its middle; and the part corresponding to the Reynolds hook has at one end a die which engages the die of the steam arm, and at its other end a roller running in the curved cam slot. This hook is really a bell-crank lever with arms that are not in the same place. The bearing on which it turns is carried on the lower end of the rocker and, therefore, is equivalent to a movable pivot similar to the hook stud of the Reynolds gear.

Fig. 81. Brown Releasing Gear for Operating Corliss Admission Valves

In the position shown, the dies are engaged. Motion of the steam rod toward the right will move the lower end of the rocker toward the left, and consequently turn the valve spindle in a right-hand direction. This will open the valve and at the same time raise the dashpot rod. Meanwhile, the roller is moving toward the left in a circular part of the cam slot, the center of which is at the center of the valve spindle. This causes the steam arm and the bell-crank lever, which has the roller at one end, to move in such a way that there is no relative motion between them. As soon, however, as the roller comes to the point where it is forced to move out of this circular path and move farther from the valve spindle, both arms of the bell-crank lever are turned downward, the dies become disen-

gaged, and the dashpot closes the valve. The slight up-and-down motion of the steam-arm die allows it to rise, while the hook die passes underneath when returning to re-engage for the next stroke. The makers claim that this gear permits a much higher speed than is possible with other Corliss gears.

Greene Gear. Another well-known drop cut-off gear is the Greene, Fig. 82. The valves are of the gridiron type, sliding on horizontal seats, the admission valves parallel to the axis of the cylinder, and the exhaust valves at right angles to the axis of the cylinder and just below it. *A A* are rock shafts turning in fixed bearings. *B B* are the admission valve stems. *C* is a slide bar, receiving a reciprocating motion from an eccentric. *T T* are tappets connected to the slide bar. They move to and fro with the slide bar and can also move independently up and down. They are made fast at their lower end to the gauge plate *D*, which slides through the guide *E*. The guide *E* is made fast to the governor rod *F* and through this means can be raised or lowered, thus regulating the height of the tappets.

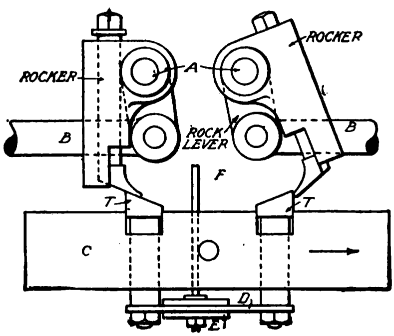


Fig. 82. Greene Drop Cut-Off Gear

As the slide bar moves toward the right, the right-hand tappet is brought into contact with the toe of the rocker, causing it to turn on its bearings and move the rock lever and the valve stem *B* toward the right, thus opening the admission valve. Since the tappet moves in a horizontal direction, while the toe of the rocker moves in an arc, it will be seen at once that they will soon become disengaged and release the valve, which is at once closed by a dashpot (not shown in the figure). If the governor raises the tappets, cut-off will be later. A nut at the bottom of the governor rod allows a proper adjustment of the guide and guage plate. As the slide bar *C* moves toward the right, the left-hand tappet comes in contact with the heel of the left-hand rocker and, both being beveled, the toe of the rocker rises in its socket, allowing the tappet to pass under. It then falls by its own weight and is ready to engage the tappet on its return and open the valve. In this gear, the disengagement of the valve throws

no load whatever on the governor, a distinct advantage over the Corliss gear. The action of the exhaust valves is not shown in the cut.

Sulzer Gear. The Sulzer gear is a drop cut-off widely used in Europe. The valves are of the poppet type, lifting straight from conical seats, so that there is no friction. They are usually placed vertically above and below the cylinder axis and are operated by eccentrics from a shaft geared to the main shaft. The admission valves are lifted from their seats by suitable levers, then released by a device equivalent in action to the Reynolds hook and are quickly closed by the action of springs.

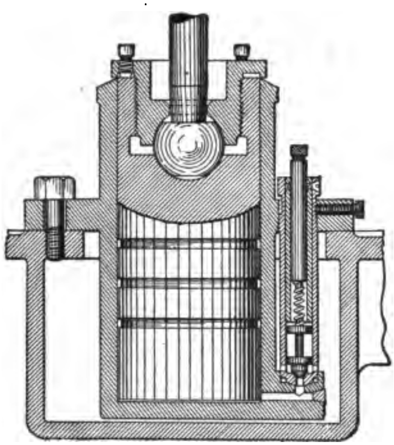


Fig. 83. Form of Vacuum Dashpot for Closing Admission Valves

The exhaust valves of all drop cut-off gears are comparatively simple in their operation, and both in opening and closing are moved by the direct action of the exhaust rods.

A common form of vacuum dashpot for closing admission valves is shown in Fig. 83. The rod coming down from the steam arm makes a ball-and-socket joint with the dashpot piston. The dashpot is often let down into the engine frame, as shown. When lifted, the piston produces a partial vacuum underneath it so that it tends to drop quickly as soon as the valve gear is released. On some of the largest modern engines where the valves are very heavy, steam-loaded dashpots are used; that is, the dashpot piston has steam pressure on one side, and an air cushion on the other prevents it from striking the bottom of the dashpot.

CORLISS VALVE SETTING

The setting of a Corliss valve gear is a much longer process than the setting of a plain slide valve, but is nevertheless a comparatively simple matter, for the various adjustments are nearly all independent of one another. In gears like that shown in Fig. 77, the length of both the eccentric rod and the carrier rod are usually adjustable, and the former should be of such length that the carrier

arm swings equal distances on each side of a vertical line through its pivot, and the carrier rod should be adjusted until the wrist plate oscillates symmetrically about a vertical line through its pivot. Nearly all Corliss engines have one mark on the wrist plate hub and three on the wrist plate stand, as shown in Fig. 84, and the wrist plate should swing so that *A*, the mark it carries, moves from *C* to *D*, but not beyond either one. When *A* is in line with *B*, the wrist plate is in mid-position. The valves are then not in their exact mid-position, but it is customary to regard them as being in mid-position, and to speak of the laps as the amount which the port is covered by the valve when the wrist plate is in mid-position.

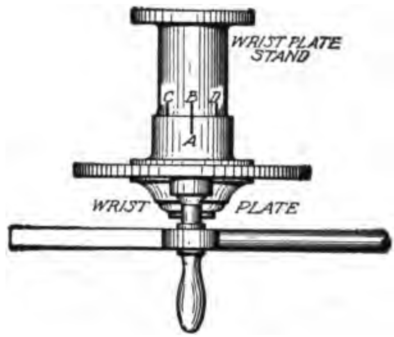


Fig. 84. Corliss Wrist Plate for Adjusting Carrier Rod

Adjusting Steam Lap. To set the valves, remove the bonnets or covers of the valve chambers on the side opposite the gear. The ends of the valves are circular, but on their inside the cross section is as shown in Fig. 85. On the end, in line with the finished edge of the valve and on the seat in line with the edge of the steam port, are marks, as shown in Fig. 85. When these marks coincide, the valve is either just opening or just closing, and when in any other position, the amount of opening or the amount by which the port is closed is shown directly by the distance between the marks. Block the wrist plate in mid-position, hook up the admission valves, and adjust the length of the steam rods by means of the right and left threads provided for the purpose, until the ports are covered by the amount of lap indicated in Table II, opposite the given size of engine.

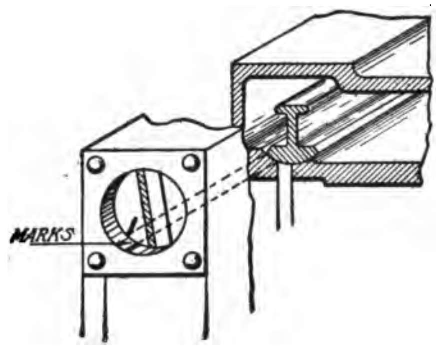


Fig. 85. Diagram Showing Method of Adjusting Steam Lap for Corliss Valves

TABLE II

Standard Lap and Clearance Values

Diameter of Cylinder Inches	Steam Lap Inches	Exhaust Clearance Inches
12	$\frac{1}{4}$	$\frac{1}{32}$
14 to 16	$\frac{5}{16}$	$\frac{1}{32}$
16 to 22	$\frac{3}{8}$	$\frac{1}{16}$
22 to 28	$\frac{7}{16}$	$\frac{3}{32}$
28 to 36	$\frac{1}{2}$	$\frac{1}{8}$
36 to 42	$\frac{5}{8}$	$\frac{5}{32}$

Adjusting Exhaust Clearance and Lead. Next adjust the exhaust rods until the exhaust ports are open an amount equal to the clearance given in Table II. Set the engine on its head-end dead point, hook the carrier rod onto the wrist plate and in the direction in which the engine is to run, turn the eccentric enough to open the head-end admission valve by a proper amount of lead; then the eccentric will be $(90+\theta)$ degrees ahead of the crank. The proper amount of lead will depend upon both the design of the gear and the speed at which the engine is to run; and may vary from $\frac{1}{32}$ inch for small engines to as much as $\frac{5}{32}$ inch or $\frac{3}{16}$ inch for large engines and those of higher speed. When the proper amount of lead has been obtained, fasten the eccentric on the shaft by means of the set screw and make sure by trial that the wrist plate moves to its extremes of travel. The dashpot rods must be adjusted so that when the dashpot piston is at its lowest position, the hooks, Fig. 78, descend just far enough for the hook dies to snap over the stud dies with about $\frac{1}{32}$ inch to $\frac{1}{16}$ inch to spare, depending on the size of the gear.

Adjusting Cut-Off. To adjust and equalize the cut-off, lift the governor to about the position that it will occupy when running at normal speed, and put a block under the collar to hold it in this position. First, set the double lever at the right of the governor cam rods, so that it makes approximately equal angles with each rod, and then turn the engine over by hand until the piston has moved to the desired point of cut-off. Adjust the proper cam rod until the knock-off cam strikes the hook and allows the valve to close, then turn the engine to the point of cut-off on the other stroke and adjust the other cam rod in a similar manner. Now set the governor

in the lowest position to which it could fall if there were no load on the engine, and set the safety cams so that in this position the hook can not open the valve. A latch is provided, on which the governor can be supported slightly above its lowest position, so that the valves can be opened by the hook when starting the engine. As soon as the engine speeds up, this latch must be moved aside, so that if the governor fails to act, it can drop to its lowest point, otherwise this latch would hold it just high enough so that the safety cams could not act.

When Corliss gears are set, as here described, the position of the eccentric may not be quite right, due to an incorrect estimate of the amount of lead required. The error is likely to produce faulty release and compression as well as poor admission, but it can not be very serious, and the engine will turn over with its own steam, so that indicator diagrams may be taken. The final adjustments can then be determined from an examination of the diagrams.

VALVE GEAR TROUBLES AND REMEDIES

Importance of Keeping Valve Gear in Condition. The valve motion, or valve gear, is primarily responsible for the correct steam distribution in all steam engines. It follows, then, that in order to maintain efficient operating conditions the valve gear should receive constant careful attention in order that any irregularities which may develop can be detected at once and the fault corrected. A great many of the different gears used in American practice are described earlier in this text. In a number of cases the methods used in adjusting the gear and setting the valve have been given. For this reason the matter presented under this heading will be principally a discussion of the methods to be followed when trouble develops under conditions of service.

Familiar Types. Of the many different types of valves gears described in the preceding pages, perhaps the most familiar ones are included under the following heads:

(1) The direct-acting duplex pump valve gear
(2) The plain D-valve or piston valve gears of the simple steam engine
(3) The Corliss engine valve gear
(4) The Stephenson link motion valve gear
(5) The Walschaert radial valve gear

DUPLEX PUMP VALVE GEAR

Description. A great variety of valve gears are used in direct-acting steam pumps. The most common form, and in many respects the most reliable, is that illustrated in Fig. 50, in the text "Steam Engines". A pump such as shown in the illustration is nothing more than two pumps combined. In this particular design the motion of the piston rod of each pump is made use of in operating the valve of the other. In such a gear the only part which is made adjustable is the length of the valve rod. It is easily seen therefore that the setting of the valve is a comparatively simple matter.

Possible Troubles. After such a pump has been in service for some time, it may be necessary to dismantle it preparatory for removal to a machine shop for repairs. An accident may happen in which one of the operating arms, which are usually made of cast iron, becomes broken. In either case the operating engineer should be in a position to readjust the parts and properly set the valves after the necessary repairs have been made.

Setting Valves. The valves of such a pump are usually of the D-type, but piston valves could be used to advantage if desired. In setting the valves the general procedure should be as follows:

Preparation. Remove the steam chest or valve chest covers so that the movement of the valve relative to the ports can be measured.

Measuring Valve Travel. Move each of the pistons as far as it will go against the head in one direction and make a pencil mark on the seat of each valve at its edge, the farthest from the center of its travel. Now move the pistons against the other cylinder heads and make pencil marks on the valve seats, but on the other edge of the valves. The marks on the valve seat indicate the travel of its valve, which should be symmetrical with the ports.

Equalizing Valve Travel. If the travel is unequal, relative to the steam ports, the valve stem should be adjusted until the valve overtravels each steam port by the same amount. When the travel of each valve has been equalized, the setting may be considered finished and the valve chest covers may be replaced and parts connected.

Variation in Conditions. In the adjustments just explained it is assumed that the gear was originally proportioned properly so as to cause the valve of each pump to open early enough to

prevent the pistons from striking the cylinder heads. It sometimes happens that the closing of the valves on the water end of the pump is such as to require a slightly different setting from that explained above in order that the pistons may be reversed to prevent striking. When such is the case the necessary adjustment should be made. Each individual case will probably need different treatment and cannot be anticipated.

PLAIN SLIDE VALVE GEAR OF SIMPLE STEAM ENGINE

Types. As previously explained the plain slide valve gear of the simple steam engine is the simplest of all steam engine valve gears. On account of its simplicity it is less liable to get out of adjustment or meet with an accident which would totally disable its action. The essential elements of this gear are shown in Figs. 3 and 4, 6 to 10, 20 and 21. It is usually found constructed in one of three forms: (1) the form in which the valve receives its motion directly from the eccentric; (2) the form in which the valve receives its motion through the medium of a rocker arm in such a manner that the valve rod and eccentric rod move in the same direction; and (3) the form in which the valve receives its motion through the medium of a rocker arm in such a manner that the valve rod and eccentric rod move in opposite directions.

Use of Rocker Arms. The use of the rockers mentioned in the forms (2) and (3) is usually made necessary by the design of the engine, which is such that the valve stem and eccentric cannot be placed so they will be in the same straight line. In the adjustment of such gears it is essential that the rocker be so located that it will vibrate equally on either side of a vertical line drawn through the fulcrum point. If the rocker is not adjusted as directed, the valve will receive a motion which may be faster in one direction than the other even though its travel is equalized. When such conditions exist it is impossible to secure a valve setting which will give a correct distribution.

Slipped Eccentric. A trouble which is sometimes experienced when the engine is in operation is caused by the eccentric becoming loose on the shaft and slipping around in such a way as to reduce the power of the engine or perhaps cause the engine to stall. This usually happens in engines where the eccentric is held

in position by means of a set screw. If a key is used the trouble is seldom experienced.

Method of Correction. When it does happen that the eccentric slips, the operating engineer can get a setting which, while not absolutely correct, will permit the engine to be operated with but little loss of time by following the directions here given:

First, set the engine on the head or crank end dead center, by inspection, with as much accuracy as is possible under the circumstances. *Second,* turn the eccentric around the shaft in the direction the engine is to run until steam will just begin to blow out at the cylinder drain cock on the end in question when the throttle is opened slightly. When this position is found, tighten the set screw in the eccentric temporarily. *Third,* turn the engine over to the other dead center and see if steam blows from the corresponding cylinder drain cock with the same degree of freedom. If it does the eccentric may be said to be in the correct position and the set screw may be securely tightened. When this is done the engine will be ready to again assume its duties. At the first opportunity the valve setting should be carefully checked by one of the methods described earlier in this text.

Increasing Power Capacity. It frequently happens in small plants using a plain slide valve engine that additional machines will be added from time to time until the engine finally becomes overloaded under ordinary conditions of operation. Under such circumstances the operator is asked to devise means of increasing the power delivered by the engine. This can be accomplished in one of the following ways: (1) by increasing the speed of the engine; (2) by increasing the pressure carried by the boiler; (3) by increasing the point of cut-off; and (4) by the combination of any two or all of the above methods.

Importance of Boiler Capacity. An examination into the methods given above reveals the fact that in every case additional load will be placed on the boiler. If the boiler capacity is sufficient to carry the additional load, then the problem can be solved, otherwise it cannot.

Increasing Speed. If the power is increased by increasing the speed of the engine to any very great degree, it will be necessary to change the size of the belt pulleys on the engine and line shaft in order not to disturb the speed of the machines.

Increasing Boiler Pressure. In increasing the power by increasing the boiler pressure, no changes are necessary unless it is thought advisable to replace any or all of the high-pressure steam pipe and fittings with extra heavy grade.

Lengthening Point of Cut-Off. If it is desired to increase the power by lengthening the point of cut-off, this can be accomplished by removing the valve and planing off the ends, thus reducing the steam lap the desired amount to give the increased cut-off. It is very essential to remove the same amount from each end of the valve, otherwise the steam lap would be different for each end. If the engine was originally cutting off at one-half stroke and it is desired to have the cut-off increased to three-fourths stroke, the amount of metal which should be removed to give the desired condition can easily be determined by drawing a Zeuner diagram from the valve in question. When the valve is finally reconstructed and placed in position in the steam or valve chest, it will be necessary to change the angle of advance of the eccentric in order to secure the proper amount of lead. To secure the proper setting it would be advisable to follow the directions given earlier in this text.

Use of Double Valve. As has been previously pointed out, the plain D-valve possesses certain objectionable features in the matter of steam distribution which is partially overcome by the use of a double valve. The Meyer valve is perhaps the most common form of double valve, a description of which is given on pages 73 to 78 of this text.

Setting Meyer Valve. In setting the Meyer valve, the main valve is set in the same manner as the ordinary simple D-valve. This main valve controls the admission, release, and compression points, while the riding, or secondary, valve controls the point of cut-off. Having correctly set the main valve, connect the riding valve to its eccentric and adjust the rods so that its travel is equal on each side of its central position, in exactly the same way as directed for the simple D-valve. When this is done, place the piston at the point where cut-off is desired and rotate the riding eccentric in the direction the engine is to run until a point is reached where the valve is just cutting off. When this point is reached fasten the riding eccentric to the shaft. Next place the

piston at the same relative position on the other stroke, and, if cut-off is just occurring, the valve may be said to be correctly set and the riding eccentric securely fastened. If cut-off does not occur at the same point on each end, make adjustments of the eccentric and valve rod until the cut-off points are equalized.

Pounding or Knocking. The question of pounding or knocking is discussed in "Steam Engines", but since this is frequently caused by improper valve setting, it seems well to give this troublesome matter a brief consideration.

Indications of Faulty Valve Action. If an annoying pound is heard which is difficult to locate, it is probably due to an improperly set valve. If this is the real cause of the trouble, it will be easily shown by indicator cards taken from the engine when under regular operating conditions. If the pound is due to valve action it will be revealed in the indicator cards in one or all of the following three things: (1) by compression beginning so early that the compression pressure exceeds the steam line pressure, thus causing the valve to be raised from its seat until the admission point is reached when the valve is forced to its seat with a "slam"; (2) by admission occurring so late that the lost motion first "runs out" and is then taken up after steam has been admitted; and (3) by the unequal distribution of power between the two ends of the cylinder, thus causing nearly all the work to be done in one end.

Correction of Fault. By following the directions as previously given for valve setting by measurement or by indicator, it becomes a comparatively small matter to correct the trouble.

CORLISS VALVE GEAR

Description. The Corliss valve gear is the most widely known of all the types of so-called "drop cut-off" valve gears. It is more economical than most other types from the standpoint of steam consumption but, on account of its peculiar construction and multiplicity of parts, is not adapted for high-speed work, say, above 100 revolutions per minute. Directions for setting a Corliss valve gear have been presented earlier in this text and need not be repeated here, but there is a word of caution which should be emphasized.

Possible Troubles. The rods connecting the steam valve arms with the dash pots should be adjusted so that when down

as far as they will go and with the wrist plate in its extremes of travel the stud die on the valve arm will just clear the shoulder on the hook die. If the rod is left too long, the steam valve stem will probably be bent, the valve arm broken, or the dash pot rod bent or broken. It may happen that the jar from the action of the dash pot will cause the dash pot rod to become loosened while in service. If this occurs the parts just mentioned may be broken in a manner similar to that when the rod is left too long in setting. Again, if the dash pot rod is left too short, the hook will not engage and, consequently, the valve will not open.

STEPHENSON VALVE GEAR

Extent of Use. The Stephenson gear, or *link motion,* as it is commonly called, is one of the oldest and best known types of reversing gears in use in the United States. For a great many years it was used almost to the exclusion of all other types of gears on American locomotives. Of recent years, however, its use has declined until today we find only comparatively few American locomotives equipped with the Stephenson reversing gear. The use of this gear is not confined entirely to locomotive service. In fact, it is made use of on steam engines in many classes of service, such as, steam tractors, steam road rollers, stationary engines, and hoisting engines.

Characteristics. *Increase of Lead in Open-Rod Construction.* One of the characteristics of the Stephenson reversing gear is that the lead of the valve increases from full to mid gear for open-rod construction and decreases from full to mid gear for crossed-rod construction. The crossed-rod construction is seldom used on engines unless service conditions are such as to make necessary its manipulation by the use of the reversing lever. The feature of increasing lead from full to mid gear, under certain conditions, is desirable on locomotives used for passenger service. In such instances the engineer will usually start the train with the reverse lever at or near the full gear position where the lead is a minimum and as the speed increases will bring the reverse lever nearer and nearer the central position where the lead is greater. This feature considered by itself is desirable since for best working conditions the lead should increase with the speed.

Back-Up Eccentric. One very desirable feature of the Stephenson gear is that it may be set to secure almost any steam distribution desirable. This is accomplished by making use of the "back-up" eccentric. Applying this method to setting the valves will, of course, disarrange the reverse, or "back-up", conditions but the "go-ahead" conditions can be almost perfectly secured.

Possible Troubles. *Lost Motion in Driving Boxes.* In the use of Stephenson gears on locomotives there is one condition which frequently arises but is rarely considered. The condition referred to is the development of lost motion in the driving boxes. In such cases, the eccentric being attached to the axle, the full amount of this lost motion is delivered to the valve with the link working in full gear. In certain other types of gears this condition would produce but very little change in the movement of the valves.

Effect of Vertical Motion of Engine. Another condition which affects the steam distribution when a Stephenson gear is used is the vertical motion of the engine on its springs caused by irregularities in the track.

Setting Valve. In setting the valve on an engine using the Stephenson gear, the fundamental principles involved are exactly the same as those given for the setting of a plain slide valve gear. We need to keep constantly in mind, however, that there are two eccentrics and two eccentric rods to deal with instead of one.

Typical Plain Slide Valve Setting. As an example let us consider the case of an engine fitted with a plain slide valve gear. Suppose it is desired to give the valve a lead of $\frac{1}{32}$ inch on both the head and crank ends. An examination of the valve discloses the fact that the lead on the head end is $\frac{1}{8}$ inch and that on the crank end is $\frac{1}{32}$ inch, which is the desired amount. The problem is to reduce the lead on the head end $\frac{3}{32}$ inch without disturbing the lead on the crank end. This problem can be solved by reducing the lead on the head end $\frac{3}{64}$ inch by changing the length of the valve rod and an additional $\frac{3}{64}$ inch by changing the angle of advance of the eccentric on the shaft. If the work is carefully done the results should show a lead of $\frac{1}{32}$ inch on both ends, unless the angularity of the eccentric rod is a very considerable amount.

Stephenson Valve Setting. Now suppose that the simple slide valve gear on this engine has been replaced by a Stephenson revers-

ing gear and that an examination of the valve with the reverse lever in full gear position shows the lead on the head end to be $\frac{1}{8}$ inch and that on the crank end $\frac{1}{32}$ inch for the forward position of the reverse lever, while for the backward position the leads on both the head and crank ends are found to be correct, namely, $\frac{1}{32}$ inch. In this case, the same as before, it is desired to secure a lead of $\frac{1}{32}$ inch on each end when the reverse lever is in both the forward and backward positions. To accomplish this with the reverse lever in the full forward position, it will be necessary to reduce the lead $\frac{3}{64}$ inch by changing the length of the eccentric rod and an additional $\frac{3}{64}$ inch by changing the position of the eccentric on the shaft. If the work is carefully done the desired results will be approximately secured.

Differences in the Two Settings. In the example just presented it should be noted that in the case of the simple gear the adjustments were made on the valve rod and eccentric, while in the case of the Stephenson gear they were made on the eccentric rod and eccentric. This correction of one-half the error on the eccentric and one-half on the eccentric rod, instead of on the valve rod, is necessary in order to permit the conditions on the reverse direction to remain unchanged. Other adjustments of a like nature can be made in a similar manner.

Variation in Conditions. Unfortunately, in practice, the Stephenson reversing gears are not always constructed so as to permit all the adjustments mentioned above. In such instances a compromise will have to be made.

WALSCHAERT GEAR

Extent of Use. The Walschaert gear has been used abroad for many years but never attained prominence in this country until ten or twelve years ago. It represents the most satisfactory type of radial reversing gear now in service in the United States. It is now being equipped on approximately 80 per cent of all new American locomotives, the remaining 20 per cent being fitted with the Stephenson gear. Its use, however, is confined almost exclusively to locomotive service.

Comparison with Stephenson Gear. One of the chief advantages of the Walschaert gear is the accessibility of all the parts and

the comparative ease with which repairs can be made. The parts of the gear being located outside, the space below the boiler may be used for other parts not so necessarily accessible. The chief point in which the Walschaert gear differs from the Stephenson gear on the action of the valve is that the former gives a constant lead for all positions of the reverse lever. Both gears are adaptable for use with any form of locomotive valve yet designed. The usual construction of the Walschaert gear is such as to permit little or no adjustments being made on the road. It is unusually free from any inclination of the parts to cause trouble through heating. Cases are known where improperly designed gears gave some trouble by the eccentric rod pins heating due to the twisting effect between the driving wheels and engine frame caused by unusual conditions of track and service. This, however, is a matter easily corrected.

Lost motion in the driving boxes produces much less effect on the motion of the valve when a Walschaert gear is used than when a Stephenson gear is employed. Neither does the up-and-down motion of the engine on its springs affect the steam distribution unless the connection of the eccentric rod to the link foot is placed at too high a point above the center line of the axle. In all well-designed Walschaert gears it is necessary that the trunnion upon which the link oscillates be fixed at an unvarying distance from the cylinder. In the fulfillment of this requirement it will be observed that the link bracket is invariably attached to the guide bearer, or yoke, and the slide for the valve stem is mounted on the upper guide bar. In some types of locomotives the construction is such that a large cast-steel bracket is laid across, joining the bars of the engine frame on both sides just back of the guide yoke, which acts as a frame binder and brace and a carrier for the link bracket. In still another type, the large casting is bolted to the guide yoke as well as the frame, thus forming a most substantial construction.

Repairs. With the Walschaert gear in service, if a break occurs within the valve gear, the difference in time consumed in making the temporary repairs necessary to get the engine moving under its own steam is greatly in its favor. This is one of the principal reasons for its adoption, since it means less time lost in delays.

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